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MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

Battlefront of Civilization

Tradition, inertia, mental laziness, ultra-conservatism, nearsighted self-interest, and a host of other enemies constantly oppose societal advance. They have to be fought. This fighting can even occasionally be of the character of actual murderous war among nationally segregated peoples, but more often it is an intellectual struggle among mentally diverse kinds of people. The outstanding struggle of the moment is that of science against ignorance and prejudice; of facts against opinions; of truth against falsehood.

VERNON KELLOGG

(From address at Sigma Xi Dinner, New York, November 17, 1924)

APRIL 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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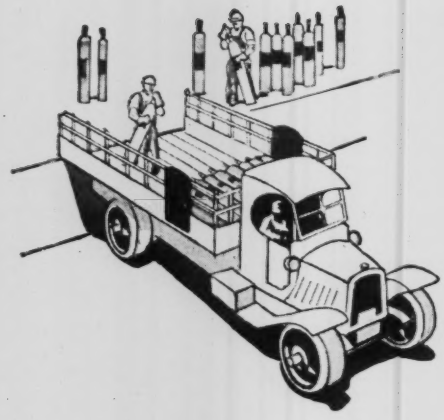

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The Monthly Journal Published by

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Volume 47 April, 1925 Number 4

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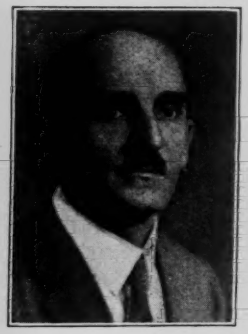
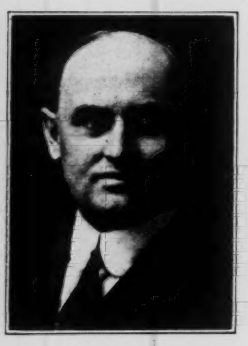


R. HILDEBRAND



L. V. LUDY

P. W. THOMPSON

E. H. TENNEY

Brunel

J. K. WOOD

Contributors to this Issue

George D. Babcock, author of Production Control, received the degree of B.S. in electrical engineering at Purdue University in 1904, and for the next three years was professor of mechanic arts at Syracuse University. In 1907 he became production manager of the H. H. Franklin Manufacturing Co., Syracuse, N. Y. He resigned from this position in 1917 to enter the army as major in the Ordnance Department, and the following year was commissioned lieutenant-colonel. He acted as supply officer of the A. E. F., directing the procurement, storage and distribution of ordnance matériel. Since 1919 Mr. Babcock has been manufacturing executive of the Holt Manufacturing Co., Peoria, Ill. He is the author of Taylor System in Franklin Management, and various articles in such technical journals as *Industrial Management, Bulletin of the Taylor Society,* etc.

* * * * *

Joseph K. Wood, author of the paper on Mechanical Springs, received his engineering education in Cooper Institute of Technology. As a consulting engineer on spring design and engineering materials Mr. Wood has been engaged by both the Westinghouse Electric & Manufacturing Co. and the Western Electric Co. He is the inventor of a flexible gear designed to eliminate chattering slip in electric locomotives and street cars. Mr. Wood served as an officer in engineering service during the War. In 1923 he was appointed chairman of the A.S.M.E. Special Research Committee on Spring Design.

* * * * *

L. H. Morrison, who writes on The Industrial Use of the Oil Engine is associate editor of *Power*. Mr. Morrison studied mechanical engineering at the University of Missouri and was then for five years connected with the English Iron Works in Kansas City. Later he served as sales engineer for the Texas Machinery Co., and as assistant supervisor of Diesel engines with the Texas Light & Power Co. During the War Mr. Morrison was in the Ordnance Department of the Army as production officer of the instrument department of Frankford Arsenal.

* * * * *

R. Hildebrand, whose paper on The Solid-Injection Oil Engine is included in this issue, is chief engineer of the Diesel department of the Fulton Iron Works. Mr. Hildebrand was born and educated in Germany, but has been a citizen of this country for the last twenty-four years. His experience covers air compressors, pumps, and steam and internal-combustion engines. He was associated with the Busch-Sulzer Brothers Diesel Engine Co. for three years before becoming connected with the Fulton Iron Works in 1914.

* * * * *

L. V. Ludy, professor of experimental engineering at Purdue University, is the author of Test of a Prosser-Type Reciprocating Steam Engine. He was graduated from Purdue University in 1898 with the degree of B.S. in mechanical engineering, and two years later received his M.E. In 1902 Professor Ludy served as an engineer in the Bureau of Forestry in the Department of Agriculture and the following year became an inspector for the Carnegie Steel Co. He then turned to teaching, and from 1903 to 1911 was instructor and professor of mechanical engineering at Purdue. For one year he held the professorship of steam and gas engineering at Wisconsin University, after which he returned to take his present position. Professor Ludy was in the Service of the United States Shipping Board Emergency Fleet Corporation during 1918 and practiced consulting engineering during the summers. He is the author of The Steam Engine, Locomotive Boilers and Engines, The Air Brake, Valve Gears, and Mechanical Engineering Laboratory Manual.

* * * * *

P. W. Thompson, who holds the position of chief assistant engineer of power plants of the Detroit Edison Co., Detroit, Mich., contributes an article on Operating Experiences of the Trenton Channel Plant of that company. Mr. Thompson was graduated from Cornell University as a mechanical engineer with the class of 1910. He worked for a year with the Bell Telephone and Western Union Telegraph Co., and then returned to the University as an instructor of steam-power engineering. Since 1913 he has been associated with the Detroit Edison Co.

* * * * *

Edward H. Tenney, author of Pulverized Coal at Cahokia, is chief engineer of power plants of the Union Electric Light & Power Co., St. Louis, Mo. He was graduated from Oberlin College in 1903 with the degree of B.A. and from Ohio State University in 1905 with the degree of M.E. He became connected with the Union Electric Light & Power Co. in 1905 as draftsman, and one year later was appointed assistant to the chief engineer of power plants, which position he held until 1919 when he became chief engineer.

* * * * *

At a session held at the A.S.M.E. Annual Meeting on December 2, 1924, two papers on turbo locomotives were presented, one of which gave particulars of the Zoelly locomotive. The other, by **George F. Jones** and **T. Laurence Hale,** which appears in this issue, describes the Ramsay Condensing Turbo-Electric Locomotive. Messrs. Jones and Hale are both associated with the firm of Thompson, Watt & Co., Ltd., London, England.

* * * * *

Thomas S. Curtis, whose paper on Refractories for Oil-Burning Boilers is abstracted in this issue, holds the position of Director of Research with the Vitrefrax Co., Los Angeles, Cal.

A.S.M.E. Spring Meeting
Milwaukee, Wis., May 18–22, 1925

All signs point toward an interesting and pleasureful Spring Meeting. The technical program is nearing completion and will be announced in the A.S.M.E. News for April 7. A feature session will be a joint meeting with the American Society of Refrigerating Engineers. Watch the current issues of the News for details.

MECHANICAL ENGINEERING

Volume 47 April, 1925 No. 4

Ramsay Condensing Turbo-Electric Locomotive

General Description of the Locomotive—Design of the Condensing Plant—Control of Power and Speed Regulation—Shop and Main-Line Testing

By GEORGE F. JONES[1] and T. LAURENCE HALE, LONDON, ENGLAND

IT MAY BE SAID that just as the cylinder, boiler, and firebox have become fundamental parts of the ordinary locomotive, so have they come to determine the degree of its thermal performance and the limit of its possible power development.

With the present-day demands for increased hauling capacity and the greater need for economy in fuel consumption emphasizing the limitations of the orthodox locomotive design, it is not surprising to find engineers contemplating the attractive possibilities offered by locomotive development along lines other than the conventional.

It is apparent that there is great room for improvement when it is considered that a locomotive of the ordinary type has a thermal efficiency of only approximately six per cent and that doubling its thermal efficiency would reduce the fuel consumption by half or, alternatively, allow a much more powerful engine to be built with the present practical limits of firebox area.

The replacing of cylinders by the steam turbine is no more of an innovation than the application of condensation which is rendered possible by its introduction.

From a thermodynamic point of view it is evident that turbo-locomotives, whatever their systems of power transmission, whether electrical or gear, largely depend for their economies in fuel consumption upon the efficiency and reliability of their condensing plant.

A study of the various experimental turbo-locomotives, however, is outside the scope of this paper, which is especially devoted to the description of one type rather than to a general survey of turbo-locomotive development.

It is to be understood that the Ramsay locomotive is regarded as an experimental one, designed and constructed for the purpose of obtaining technical data and demonstrating the capabilities of the type. It conforms to British standard practice as regards the restrictions imposed by the load-line gage, etc. although naturally, in view of the radical departure in design, it has been confined mainly to test and demonstration running rather than to every-day service. It may be mentioned that compact and uniform designs were disregarded in the interests of experiment, the motor-driven auxiliaries in every case having been purposely kept separate and ungrouped from the main power plant in order to obtain readily their particular power consumptions. In addition many power-registering devices, gages, and instruments were fitted, greatly in excess of those necessary to the ordinary equipment, all of which tend to give the locomotive, and especially the cab, an intricate and complicated appearance.

General Description of the Ramsay Locomotive

The external appearance of the locomotive is shown in Fig. 1. Figs. 2 and 3 show the general disposition of the units and, as can be seen, the locomotive consists of two portions, the front and rear, connected by a special form of universal joint.

The front portion incorporates the boiler, forced-draft set, and driver's cab, the main turbo-alternator set and the auxiliary d.c. turbo-generator being situated under the boiler panel.

The rear portion of the locomotive is practically devoted to the condensing plant with axial fan, condensate extracting pump, etc.; room, however, has been found for the main water tank and coal bunker. Communication between the turbines and condenser is by way of the 24-in.-diameter exhaust pipe which is provided with a flexible rubber connection reinforced by internal rings of aluminum.

Each portion of the locomotive carries two driving motors, each pair of motors being bolted to a center cross-stretcher which carries a transmitting shaft and spur wheels. Pinions are keyed to the motor shafts and these gear with the spur wheels, the power being finally transmitted through coupling rods from the spur wheels to the driving wheels in the ordinary manner. Fig. 4 shows the spur wheel and pinion shafts, the pinions having been removed. The pinions and spur wheels are enclosed in a gear case and run in an oil bath, the gear ratio being 2.8 to 1.

The main turbine is of the impulse type and contains nine stages, the mean blade diameter being 36 in. It is designed for a steam pressure of 200 lb. per sq. in. and the steam is superheated to a total temperature of 685 deg. fahr., exhausting to a vacuum of 27 in. The turbine is flexibly coupled to a three-phase alternator and has a speed range of from 1800 r.p.m. at starting to 3600 r.p.m. at 60 m.p.h. The three-phase alternator is designed to develop 890 kw., at a maximum pressure of 600 volts. Fig. 5 shows the main turbo set in course of erection in the locomotive frames at the Scotswood works of Sir W. G. Armstrong, Whitworth & Co., Ltd.

The auxiliary turbine is a single-stage machine flexibly coupled to a d.c. generator which provides the energy for the excitation of the main alternator poles as well as for the auxiliary d.c. motors driving the condenser fan, condenser rotor, condenser extracting pump, and the forced-draft set for the boiler. It also supplies the necessary current for lighting the train. The auxiliary turbine operates under the same steam conditions as apply to the main turbo set.

The boiler is of the ordinary locomotive type and the combustion chamber is supplied with air by a high-speed forced-draft set situated in the locomotive cab. A simple locking device is provided on the furnace door to prevent a "blowback," the door being prevented from opening by a safety catch when the forced-draft fan is in operation.

The transmission of power from the main turbine to the driving motors is three-phase, three-phase current being supplied from the alternator to the four alternating-current slip-ring motors. Each motor has a continuous output capacity of 275 b.hp. and a one-hour rating of 360 b.hp.

The following are the tractive efforts at the rims of the driving wheels for the acceleration period from rest to 60 m.p.h.

Miles per Hour	Tractive Effort, Lb.
Starting	22,000
15	22,000
30	11,050
60	8,600
Normal running (60 m.p.h.)	6,000

Design and Description of the Condensing Plant

As the success of a locomotive of this description depends so largely upon the design and action of its condenser, it is perhaps not irrelevant to review briefly the considerations which led to the adoption of the principle of evaporation as the basis of design for the present condenser.

[1] Thompson, Watt & Co., Ltd., 9–11 Eagle St., Southampton Row, London, W. C. 1.

Contributed by the Railroad Division and presented at the Annual Meeting, New York, December 1 to 4, 1924, of The American Society of Mechanical Engineers. All papers are subject to revision.

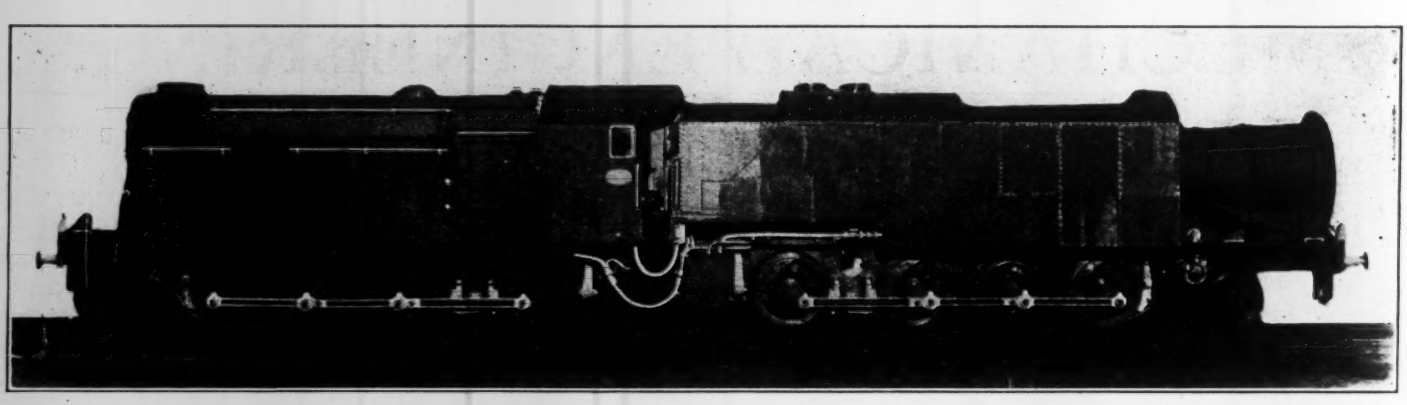


FIG. 1 EXTERNAL APPEARANCE OF THE RAMSAY TURBO-ELECTRIC LOCOMOTIVE

In the light of previous experience of the difficulties of "recooling" condensing systems, it was decided to attack the exhaust steam by direct means. It was fully realized that the method alternative to evaporation, that of direct cooling by air alone, involved employment of large condensing surfaces and excessive fan power, and that such a type of condenser would depend for its success largely upon atmospheric conditions; and, while probably producing a satisfactory vacuum in low-temperature countries, the system would be at a great disadvantage and far from satisfactory in hotter climates, such as India, etc.

Attention was therefore turned toward the evolving of a suitable condenser on the direct cooling system based on the evaporative principle as being the system that would successfully meet the atmospheric conditions of all countries.

In design the ordinary evaporative condenser has progressed but little and still remains practically as originally conceived. Such test results as existed were unreliable, and it was therefore necessary to carry out a series of preliminary experiments on small-scale condensing apparatus with fans to produce a current of air and so intensify the evaporative effect. The importance of maintaining a thin and unbroken water film upon the condensing surfaces was easily demonstrated and, of the numerous methods and contrivances tested with a view to producing satisfactory water filming of the surfaces under a blast of air, the simple method of completely immersing the surfaces in water before exposing them to the air current was found to give the most reliable and proper film effect. Repeated tests on a small condensing plant proved the type to be exceedingly economical in water consumption, in practice only approximately one pound of water being evaporated to condense one pound of steam. The fan power required was reasonably low, while the rate of heat transmission was found to be moderately high and sufficient to bring the condensing surface well within the practical limits.

Further experiments were carried out on a plant containing some 400 sq. ft. of surface. For the sake of simplicity, the condensing surfaces were composed of ordinary brass condenser tubes, and, as a natural development to facilitate the process of filming, they were arranged in the form of a circular cage or hollow drum. This drum, supported between bearings, was rotated in a tank containing water, the tubes passing through the water as they rotated.

The condenser was supplied with air by an axial fan and was tested in conjunction with a steam regenerator and the necessary wet-air pump, water filming of the surfaces and resultant evaporation taking place successfully. The promising results from this experimental work led to the design and construction of the present locomotive condenser.

GENERAL ARRANGEMENT OF CONDENSER AND ITS AUXILIARIES

Reference to Fig. 3 shows the general arrangement of the rear portion of the locomotive, and Fig. 6 gives the disposition of the condenser and its auxiliaries. Ordinary condenser practice was observed in the construction of the drum, which consists of an annular nest of standard $^3/_4$-in.-diameter brass condenser tubes ferruled into two headers. The exhaust steam entering one header is condensed, and the resultant condensate is collected and drawn off from the other header and returned to the hotwell by the rotary extraction pump as hot and clean feed for the boiler. The trunnions of the respective headers are fitted with airtight glands of the steam-sealed self-adjusting type, and this design of gland has given every satisfaction in resisting air leakage into the condenser.

As will be seen, a small motor, driving through a spur gear, turns the drum which is housed in a sheet-metal casing, the bottom of which forms a tank containing the water for purposes of filming, and fed from the main tank; a pneumatic float maintains the water at a constant level. Fig. 7 shows the robust construction of the condenser fan, which, it will be noticed, is of the axial type with guide blades.

The condensate extracting pump is of the ordinary rotary condenser design, while the air pump is of the steam-ejector type, has two stages, and is supplied with steam from the boiler through a reducing valve.

Before being erected on the locomotive, the complete condensing plant was fully tested at St. Andrews Cross Power Station, Glasgow, in conjunction with a 1500-hp. Willans triple-expansion engine, and satisfactory results were obtained.

CONTROL OF POWER AND SPEED REGULATION

As will be seen, the starting torque of the motors is about three times that of the normal and is obtained as follows:

Before starting the locomotive the auxiliary turbine is run up to speed, 3000 r.p.m., thus providing excitation for the main alternator and energy for the motor-driven auxiliaries. The main turbine set is then run up to half-speed, or 1800 r.p.m. At this period the motors are connected in cascade by the master controller located in the driver's cab. It is then that the locomotive may be started.

It is well known that when a turbine speeds up from rest to full speed its torque decreases in the ratio of two to one, and when passing through the period of half-speed its torque is one and a half times the normal, which is the torque of the main locomotive turbine when running at starting speed, namely, 1800 r.p.m. Again, two alternating-current motors when connected in cascade and running at half the speed of the alternator have, as is known, twice the turning moment that they have when connected in parallel with the same power consumption. Therefore, with the driving motors in cascade and the main turbine running at half-speed, the torque from rest to quarter-speed will be 2×1.5 or three times the normal torque.

The motors are now connected in parallel, the turbine still running at half-speed, and the speed of the locomotive increases from one-quarter to one-half speed; the torque is then 1.5 times the normal.

The turbine speed is then increased to the maximum and the torque drops from 1.5 times the normal to normal.

In controlling the speed of the Ramsay turbine locomotive from rest to 60 miles per hour, the following sequence of operations is observed:

After the main and auxiliary turbines have been brought to speed by means of the handwheels in the cab which control the steam inlets, all control of the locomotive is carried out electrically by means of the master controller in the cab.

To start the locomotive the controller wheel is moved to the first notch, thereby closing the excitation circuit of the alternator and connecting the driving motors in cascade with resistance wholly

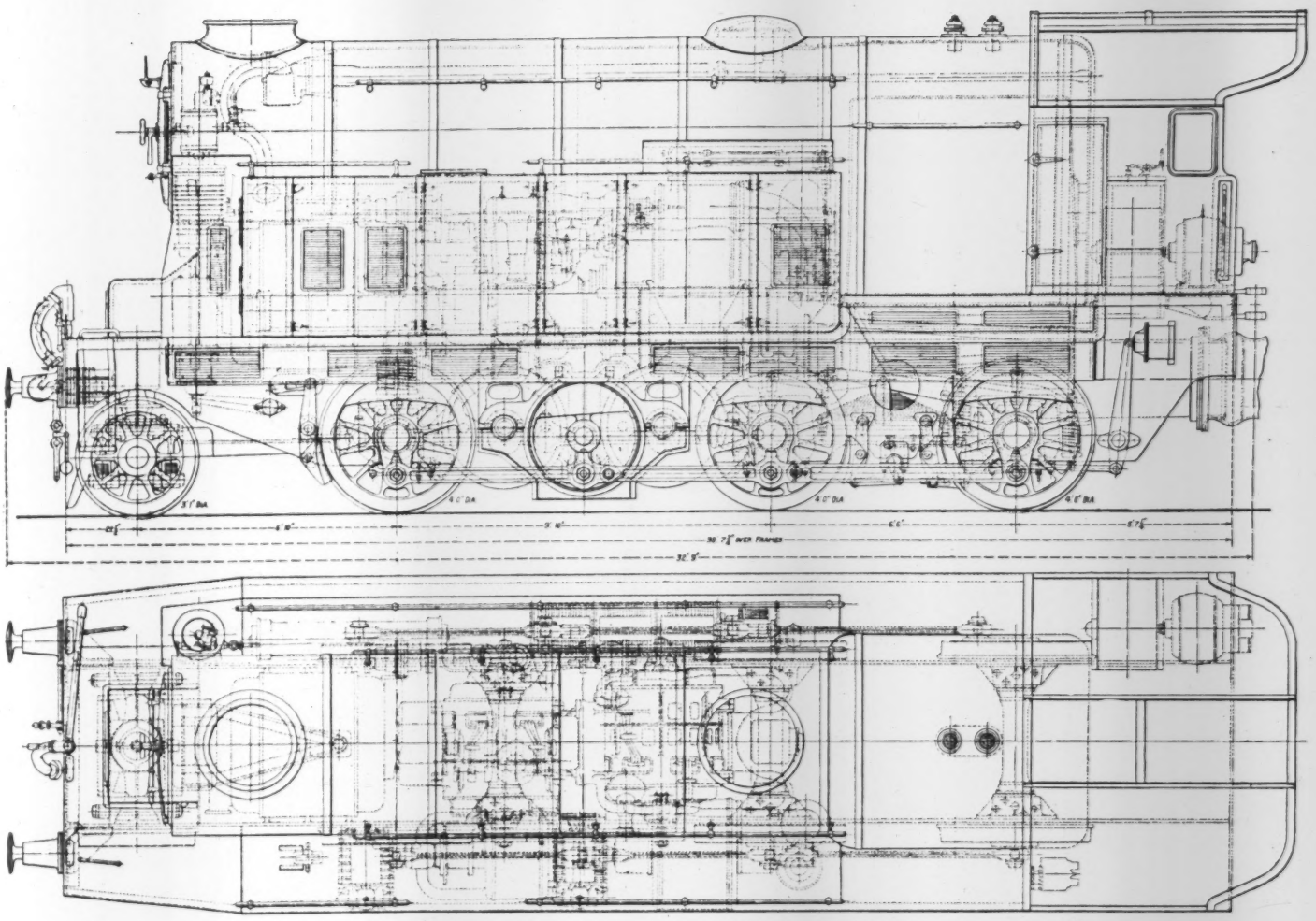


FIG. 2 GENERAL ARRANGEMENT (PLAN AND ELEVATION) OF FRONT ENGINE

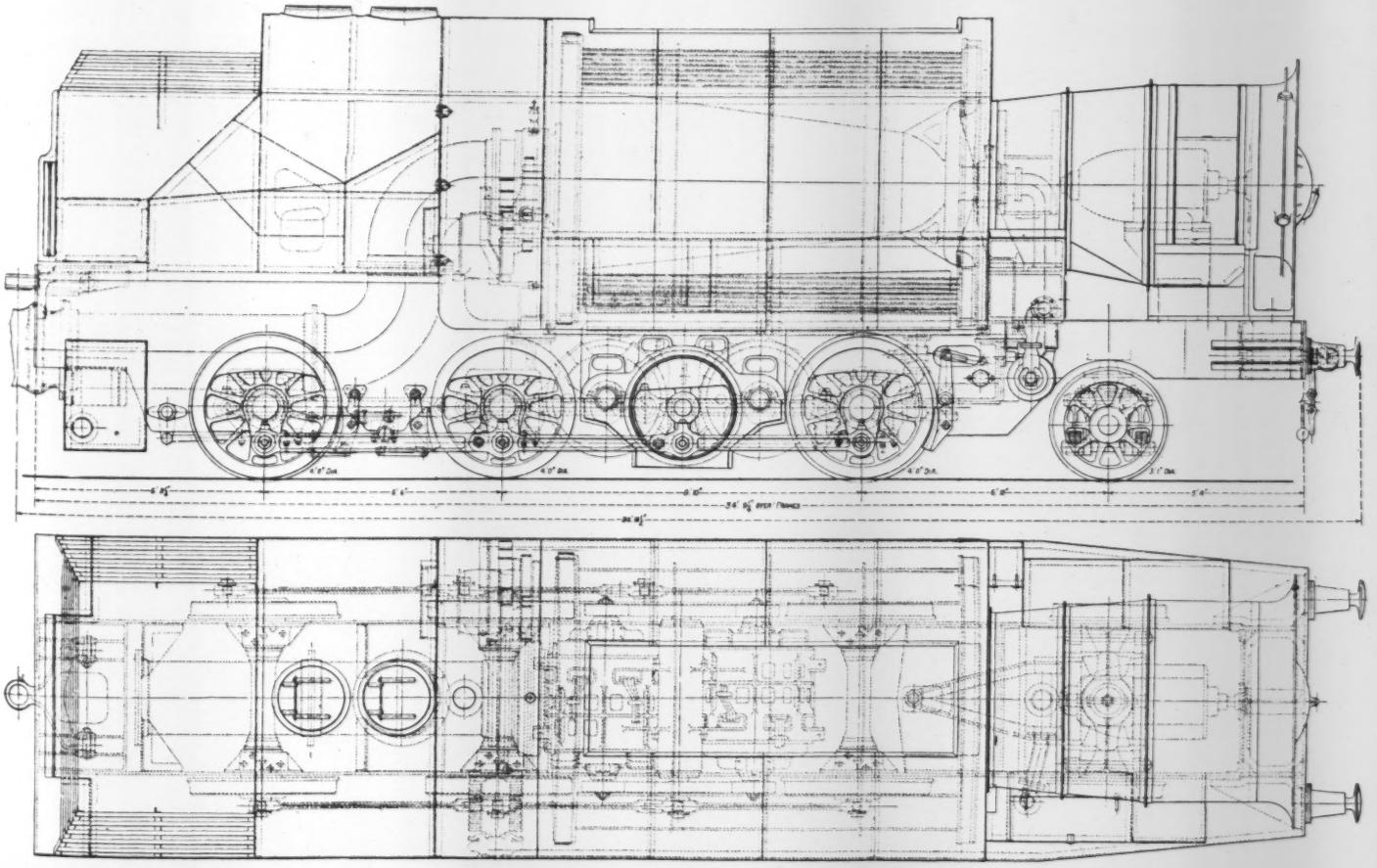


FIG. 3 GENERAL ARRANGEMENT (PLAN AND ELEVATION) OF REAR ENGINE

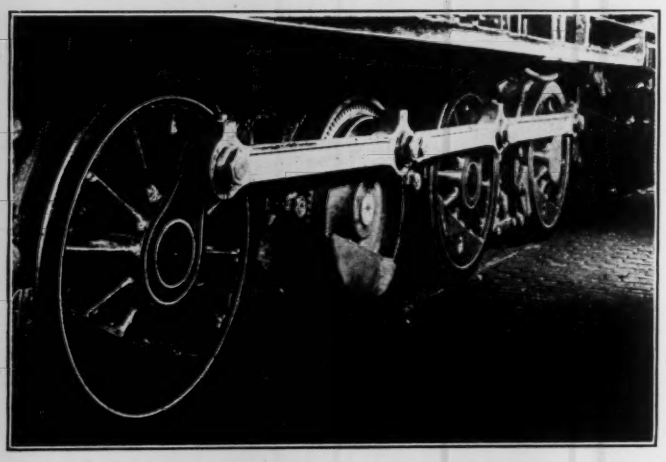


FIG. 4 DRIVING MECHANISM. SPUR WHEELS ARE DRIVEN BY PINIONS (NOT SHOWN) KEYED TO DRIVING MOTOR SHAFTS

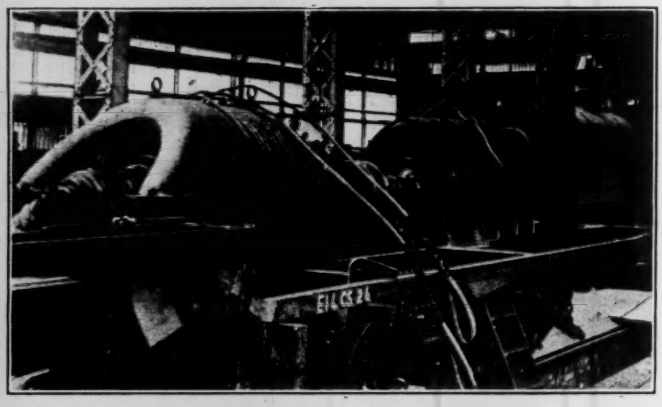


FIG. 5 MAIN TURBO SET IN COURSE OF ERECTION IN THE LOCOMOTIVE FRAMES

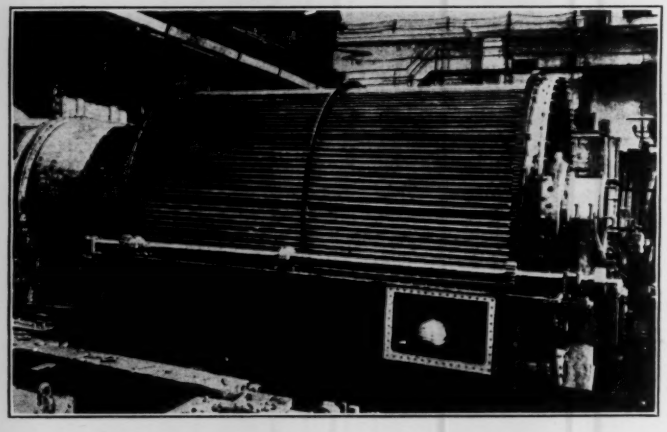


FIG. 6 DISPOSITION OF CONDENSER AND AUXILIARIES

in circuit. Further movement of the control wheel cuts out resistance until a first running position of 15 miles per hour is obtained. If a greater speed is desired, the controller wheel is moved around further and changes the motor connections from cascade to parallel. The excitation circuit being again closed, the motors once more operate under current, resistance being cut out until a second free running speed of 30 miles per hour is reached.

Further increases of speed are obtained by further moving the controller wheel around, step by step, whereby the setting of the main turbine governor is correspondingly altered, that is, the speed of the turbo-alternator is increased and thereby the periodicity. In this manner the speed of the locomotive can be increased from 30 to 60 m.p.h. A reduction in the speed of the locomotive is obtained by turning the controller wheel quickly back to the zero position, thus observing the succession of operations reverse to those of starting up to full speed. The excitation circuit being thereby opened, the motors are without current and ready for starting up again in cascade. Fig. 8 shows the arrangement of the master controller, wheel, etc.

Reversing the locomotive is of course effected by reversing the driving motors in the ordinary way.

Figs. 2 and 9 show the general arrangement of the front portion of the locomotive. The resistances, which are of the grid type, are arranged under the running plates on either side. Both vacuum and Westinghouse brakes are fitted. Fig. 10 shows the rear view of the locomotive and condenser fan.

The leading particulars of the locomotive are as follows:

Length overall	69 ft. 7 in.
Total wheelbase	59 ft. 4 in.
Rigid wheelbase	16 ft. 4 in.
Driving-wheel diameter	4 ft.
Height from rails to center line of boiler	10 ft. 3 in.
Maximum width	9 ft.
Total heating surface of boiler	1543 sq. ft.
Grate area	28 sq. ft
Tractive force at starting	22,000 lb.

SHOP AND MAIN-LINE TESTING

The trials to which the Ramsay locomotive has been subjected have extended in one form or another over a prolonged period, and often under adverse atmospheric conditions.

Shop tests to ascertain the reliability of the special electrical control of the locomotive were carried out at the works of Sir W. G. Armstrong, Whitworth & Co. before the locomotive proceeded to the Horwich works of the London, Midland & Scottish Railway for actual running trials. Disconnecting the terminals of the alternator and connecting it to water resistance made it possible to form

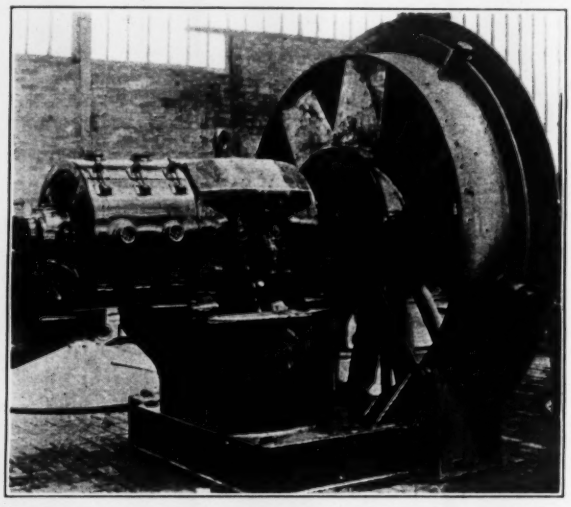


FIG. 7 CONSTRUCTION OF THE CONDENSER FAN, WHICH IS OF THE AXIAL TYPE WITH GUIDE BLADES

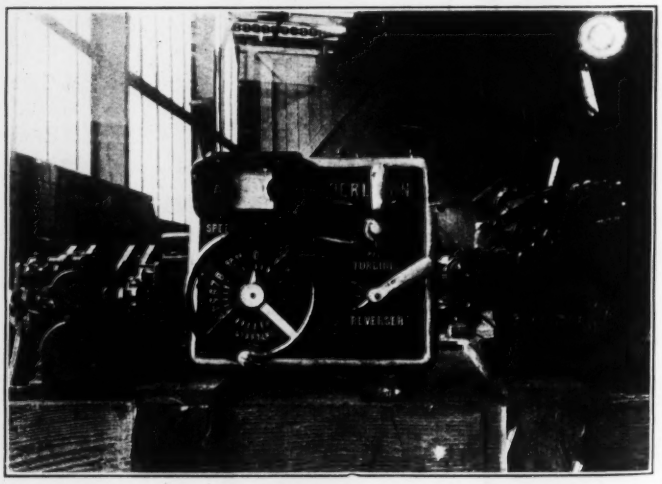


FIG. 8 ARRANGEMENT OF MASTER CONTROLLER AND WHEEL

many tests which did not justify a main-line trial, and afforded a very convenient means of obtaining readings on the condenser which would otherwise have been difficult to procure with the locomotive in motion. Numerous tests were made under these conditions and a large amount of technical data collected.

In addition several main-line trials have been accomplished. On these occasions the locomotive worked heavy trains without difficulty and a vacuum ranging from 80 to 95 per cent was regularly attained and held. Smoothness of running was most pronounced, due to the constant and even turning moment exerted by the driving motors.

It may be said that during these trials no breakdown occurred nor was it necessary to call on the pilot engine accompanying for aid. Fig. 11 shows in curve form results obtained during one of the trips of the locomotive. The lower curve shows the speed in m.p.h.; the middle, the vacuum obtained in inches of mercury; and the top curve, the boiler pressure in lb. per sq. in.

At no time during the run did the temperature of the condenser exceed 135 deg. fahr., which corresponds to a vacuum of 25 in. of mercury. Improvement in the vacuum during easier periods of the run will be noted. The work of driving and firing was carried out by ordinary railway employees, and, as all the work in speeding up the locomotive is performed electrically, the duties of the driver,

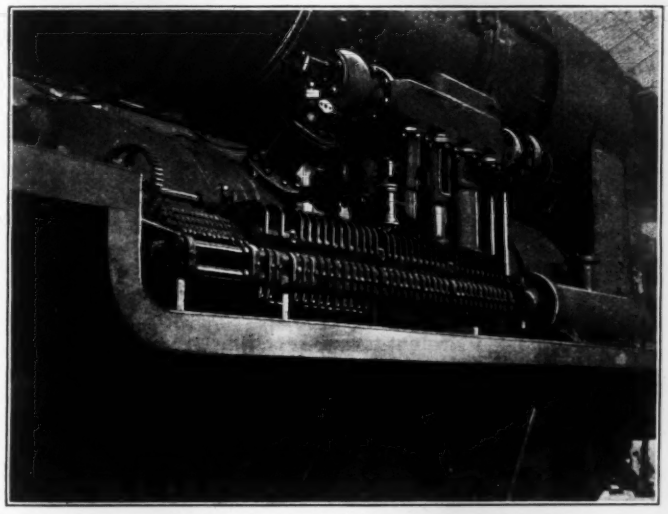


FIG. 9 RESISTANCES OF GRID TYPE ARE ARRANGED UNDER RUNNING PLATES ON EITHER SIDE

so far as controlling the locomotive is concerned, are confined to the turning of the controller wheel—the operation of which has already been described—while the duties of firing are not as onerous as those on the ordinary locomotive.

EFFICIENCY OF CONDENSING PLANT

In considering the efficiency of the condensing plant some regard must be paid to the amount of power consumed in its operation, and as in this case the major portion of the cooling is due to evaporation, the power absorbed by the fan is proportionately low.

The average power consumption for the three condenser auxiliary motors ranged between 25 b.hp. at low loads and 40 b.hp. at high loads. The heat from the air-ejector steam exhaust was recovered in the hotwell or locomotive feed tank.

No difficulties were experienced on account of deposits of scale and, although the filming water was far from pure, the slight scale which formed on the tubes assisted rather than impeded the water filming. Inspection of the inside of the tubes and headers showed them to be clean and free from deposits or corrosion after some months' running.

The foregoing is but a brief description of an engine which has proved beyond doubt that the adaptation of the turbine and condenser to the locomotive is rational and reliable.

The transmission of energy from turbine to locomotive wheels may be either carried out by mechanical gears interposed between the turbine and wheels, or electrical means may be employed.

For moderate powers there is much to be said in favor of the former, but for large powers the preponderance of the advantages lies with the latter means.

Although for service engines the power of the experimental engine would not be sufficient to warrant the employment of electrical transmission, for purposes of research the adoption was justifiable.

The exhaustive tests taken over a long period have demonstrated conclusively that there is now little difficulty in providing condens-

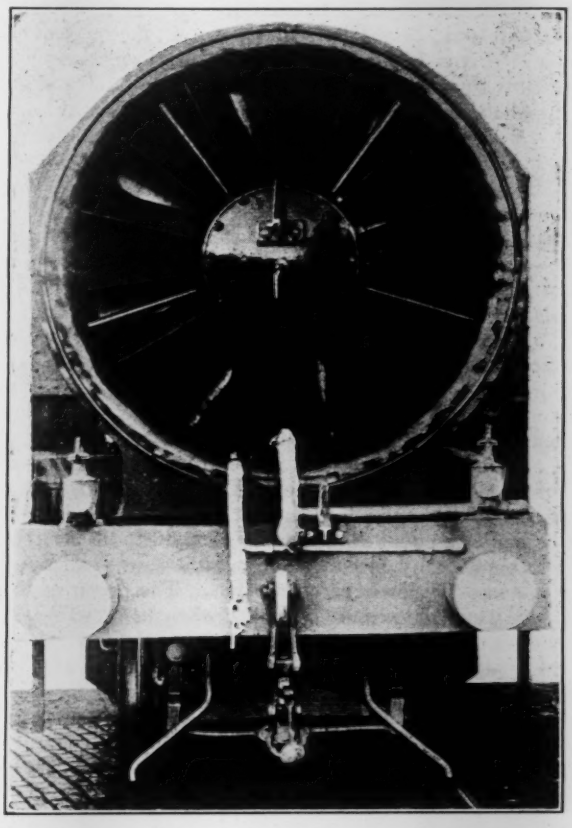


FIG. 10 REAR VIEW OF LOCOMOTIVE AND CONDENSER FAN

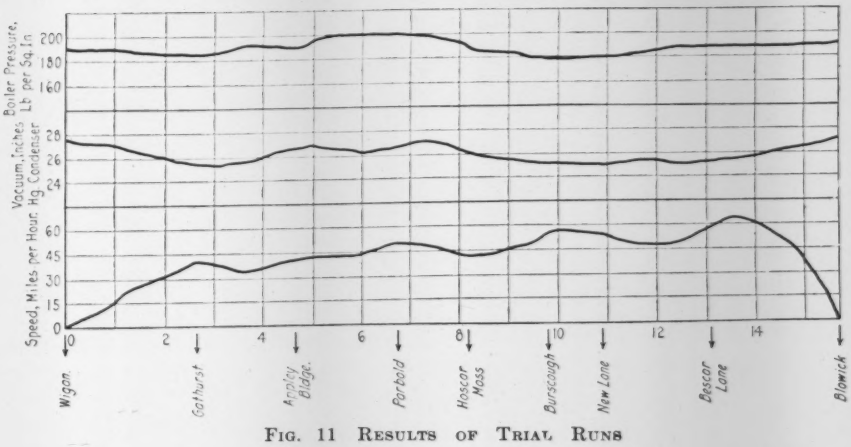


FIG. 11 RESULTS OF TRIAL RUNS

ing apparatus which will maintain the requisite economical vacuum when dealing with the heaviest of traffic.

With the elimination of all mechanisms and auxiliaries which are of necessity only for investigation and by rearranging the machinery to meet every-day running conditions, the designs of the service engines which have been developed from the knowledge and data obtained from the tests of the experimental engine become simple in construction and should show very substantial financial running economies.

Discussion of Papers on Turbo-Locomotives[1]

G. B. WARREN[2] in a written discussion gave it as his opinion that a condensing steam locomotive was the next logical development in locomotive work, and that this meant a turbine on the low-pressure end of the cycle but not necessarily on the high-pressure end. He thought the best solution was to replace the tender of a reciprocating locomotive of the present type with a car carrying not only the fuel and water, but also a low-pressure turbine driven by exhaust steam from the reciprocating engine and condensing equipment with either mechanical or electrical drive. Preliminary study indicated that the most advantageous combination would be that of a direct-current motor geared to the wheels and driven from a low-pressure turbo-generator. Such a low-pressure turbine locomotive would be about 50 to 56 ft. in length, and the combined wheelbase of a 380,000-lb. Mikado locomotive and connected low-pressure turbine locomotive would be less than 95 ft. The weight of the total combination would be about 100 tons more than that of the present locomotive and tender.

The cylinders of the reciprocating locomotive would exhaust into the pipe leading back to the low-pressure turbo-generator, and the draft would be supplied by an electrically driven blower. If desirable, the auxiliaries could be electrically operated from motors driven by a low-pressure auxiliary turbo-generator whose turbine steam flow was either in series or parallel with the main low-pressure unit.

Such a low-pressure unit would have a drawbar pull at speeds of 10 m.p.h. and above, when operating on the exhaust steam of the engine, of from 75 to 85 per cent of that of the present locomotive, whose pulling ability would be practically unaffected by the presence of the low-pressure locomotive. The starting drawbar pull could be made to equal or exceed, by from 10 to 25 per cent, that of the locomotive by bleeding a comparatively small amount of live steam into the low-pressure turbine. The power of the combined unit would be around 5000 hp. when used with a Mikado engine of 2500 to 3000 hp. That was to say, the hauling capacity of a present-type locomotive could be practically doubled without increasing the fuel consumed or the crew required for operation, and the train loadings or speed could be increased a corresponding amount.

Auxiliaries would be driven by the exhaust steam and would fluctuate in speed with the load of the engine without the attention of the engineer. The condenser would return condensed water to the boiler and should result in a considerable reduction in boiler maintenance.

The maintenance of the low-pressure turbo-electric locomotive should be very low, and the cost would probably not exceed very much that of the reciprocating steam locomotive with which it was used. It should be considerably less than that of a complete turbine locomotive whose power was equal to the power of the combined low-pressure turbine and the reciprocating locomotive. In the case of electric drives the driving system already in service could be utilized.

The indicated Rankine-cycle efficiency of the modern superheated reciprocating steam locomotive cylinder, wrote Mr. Warren, was in the neighborhood of 75 per cent at 50 to 45 per cent cut-off, and most of the loss was in the pressure release when the exhaust valve opened. Part of this was recoverable by the low-pressure turbine system. The mechanical efficiency in transmitting the indicated power to the crankpin must be exceedingly high. No turbine system at present available could do more than equal this efficiency on the high-pressure-to-atmosphere part of the cycle in the capacities used on locomotives and with losses in transmission allowed. It was also certain that no high-pressure non-condensing turbine system approached in simplicity and ruggedness that of the cylinders, valve gear, and side rods of the reciprocating locomotive. As boiler pressures were increased the difficulties of using a high-pressure turbine would increase, while those for the reciprocating engine would be unaffected.

With the low-pressure turbine system the difficulties of the transmission of power to the wheels would be cut in half. This was an item of some importance when locomotives of 5000 hp. were considered. The starting characteristics of the combination were more favorable than those of a steam locomotive. This would not be true for the geared or alternating-current electric-drive turbine locomotive.

The low-pressure turbine would have larger clearances, fewer stages, longer buckets, greater ruggedness, higher efficiency, and much better sustained efficiency than an initial high-pressure turbine.

As regarded the weight efficiency of the turbine locomotive, if it was assumed that its tender, when loaded, weighed 100,000 lb. and the locomotive 130,000 lb., and the maximum power was assumed as 850 hp., then the weight including the tender would be 271 lb. per hp., and excluding the tender, 153 lb. per hp.

The weight of the American Locomotive Company's H-10 Mikado locomotive was 535,000 lb., with loaded tender, and 350,000 lb. without tender. The maximum power was approximately 2700 hp., making the weight 198 lb. per hp. with tender, and 124 lb. per hp. without tender.

The Pennsylvania Railroad's L-1s Mikado engine weighed 500,000 lb. with loaded tender, and 315,000 lb. without tender, and developed approximately 2500 hp. maximum. This made 200 lb. per hp. with tender, and 126 lb. per hp. without tender.

The No. 5000, 3-cylinder Mountain-type American Locomotive Company locomotive weighed 570,000 lb. with loaded tender and 390,000 lb. without tender, and developed approximately 2700 hp. This gave a weight of 211 lb. per hp. with tender, and 136 lb. per hp. without tender.

It could thus be seen that the Zoelly locomotive with the assumed tender weight was only 35 per cent heavier per horsepower than the best American practice and without tender was only 17 per cent heavier per horsepower. This was on the basis of a comparison involving American locomotives of three times the capacity. This was not a fair comparison, inasmuch as the weight efficiency should be better as the capacity of the machine increased, other conditions being equal.

The proposed combination of a Midako reciprocating locomotive and a low-pressure turbine-driven condensing locomotive would have a weight of approximately 820,000 lb. when loaded with water and coal, and would have a mixiumum power of about 4500 hp. This would make a weight of 183 lb. per hp., which compared very favorably with the best American practice including the tender. It did not follow, therefore, that a condensing turbine locomotive had to be considerably heavier than a standard non-condensing locomotive. If properly designed, the reduction in boiler capacity and water and fuel storage per horsepower should offset, to a very great extent, the weight of condensing equipment required.

In a written discussion R. Eksergian,[1] while admitting the possibility of thermodynamic advantage with lower back pressure, the impossibility of using even a moderate vacuum in a reciprocating engine and hence the inevitability of the turbine, pointed out that drastic efforts in increasing thermal efficiencies were not likely to be effective unless they reacted in increased weight efficiency—defined as the maximum hauling capacity of a locomotive limited by the fixed weight limitations of the roadway, bridges, etc. The mechanism must transmit large torque loadings at low speeds, and the power plant must have a minimum weight to give the maximum tractive force at the higher speeds for a given axle, as well as total, weight of the locomotive. It was thus evident that high thermal economies did not necessarily imply satisfactory locomotive performance.

Assuming a satisfactory condenser, and considering the torque characteristics and steam consumption of the turbine as a variable-speed motor, let

T = torque developed by a turbine-stage element, lb-ft.

C_1 = absolute velocity of jet at entrance to movable blade, ft. per sec.

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[1] Discussion of papers on the Ramsay and Zoelly turbo-locomotives presented at the A.S.M.E. Annual Meeting, New York, December 1 to 4, 1924. Mr. Zoelly's paper describing his locomotive was published in MECHANICAL ENGINEERING, November, 1924, p. 653.

[2] Turbine Engineering Department, General Electric Co., Schenectady, N. Y.

[1] Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

ϕ = ratio of relative velocity at exit to that at entrance, ft. per sec.
U = peripheral velocity of blade, ft. per sec.
R = mean radius of blade, ft.
w = weight of steam per second, lb. per sec.
α_1 = angle of jet to entrance of movable blade.

Impulse Type	*Reaction Type*
$T = \frac{W}{g} RC_1(1+\phi)\left(\cos\alpha_1 - \frac{U}{C_1}\right)$	$T = \frac{W}{g} RC_1\left(2\cos\alpha_1 - \frac{U}{C_1}\right)$

For maximum efficiency,

$\frac{U_m}{C_1} = \frac{\cos\alpha_1}{2}$	$\frac{U_m}{C_1} = \cos\alpha_1$

Then rated torque

$T_m = \frac{W}{g} RC_1(1+\phi)\frac{\cos\alpha_1}{2}$	$T_m = \frac{W}{g} RC_1\cos\alpha_1$
$T = 2T_m \frac{1+\phi}{1+\phi_m}\frac{\cos\alpha_1 - U/C_1}{\cos\alpha_1}$	$T = T_m \frac{2\cos\alpha_1 - U/C_1}{\cos\alpha_1}$

Hence when $U = 0$, $T = T_s$

$T_s = 2T_m \frac{1+\phi}{1+\phi_m}$	$T_s = 2T_m$

For the variation of torque of either type,

$$T = T_s\left(1 - \frac{U}{C_1\cos\alpha_1}\right) = T_s - KU \quad \text{(straight-line variation)}$$

and assuming $\phi_s = \phi_m$ approximately, then

$$T_s = 2T_m$$

That was, the initial torque for the same steam consumption corresponding to the torque at rated speed was approximately double the torque at rated speed.

In a reciprocating locomotive, continued Mr. Eksergian, maximum horsepower was obtained around 1000 ft. per min. piston speed in 200 r.p.m. of the drivers, and the tractive force at this speed might be taken approximately for a road engine as 0.43 of the initial tractive force. That was, the ratio of the initial to the rated tractive force was 2.3 to 1.

Further investigation showed clearly that the torque characteristics of the turbine compared favorably with those of the reciprocating steam locomotive.

The steam consumption of a reciprocating locomotive, the draft, and the consequent coal consumption all increased with the speed up to the limit of the adhesion range, say, at 80 r.p.m. Above this the steam consumption decreased to its minimum value around 200 r.p.m. along the horsepower range.

With the turbine locomotive, full steam blast—i.e., the steam consumption per hour at rated horsepower—or more was required during the adhesion range, say, to 80 r.p.m. of the drivers. Thus at the very low speeds the turbine locomotive was not as economical as the reciprocating locomotive. In the case of the 50 per cent cut-off locomotive for slow drag service this economy in favor of the reciprocating locomotive was more emphatic. However, with a change in gear ratio for freight service and considering the very frequent small accelerating period for passenger locomotives, the loss in excessive steam consumption with the turbine at the low speeds was not so excessive as might at first be supposed. At the higher revolutions the turbine locomotive was of course superior to the reciprocating steam locomotive.

The transference of rotating motion to reciprocating motion by means of a jackshaft, etc. and side rods was not a simple problem. For a given adhesion weight, the rod loads, axles, etc. were considerably in excess of ordinary reciprocating-steam-locomotive proportions. It might be stated very emphatically that 180,000 lb. adhesive load was the extreme limit of a single jackshaft drive, whereas comfortable proportions might be limited to 150,000 lb. adhesive load per jackshaft. The adhesive loading on modern American locomotives ranged from 180,000 to 240,000 lb. for passenger service and from 240,000 to 300,000 lb. and over for freight service, showing the necessity, in a possible adoption of the turbine locomotive to our large motive power, for the use of two separate units.

As a concrete example of the difficulties in the design of the machinery, the jackshaft driving only three pairs of drivers with axle loadings of 60,000 lb., i.e., 180,000 lb. adhesive weight, could be considered.

Assuming a driver 72 in. in diameter with a 16-in. crank throw, a limiting value of stroke to driver diameter for wheel-center proportions might be taken at 0.45. Then, assuming ample torsional flexibility at the jackshaft in the form of a flexible gear or pinion, etc. the maximum side-rod load would be

$$K = \frac{0.33 \times 60{,}000 \times 3 \times 36 \times 1.41}{16} = 190{,}000 \text{ lb.}$$

and with poor alignment this might reach a value as high as 270,000 lb. For the bearing value on the pin, the side-rod load would be taken at $\frac{190{,}000}{1.41} = 134{,}500$ lb.

Pin diameter $= \sqrt{\frac{134{,}500}{1{,}500}} = 9.5$ in. Use $9\frac{1}{2}$ by $9\frac{1}{2}$-in. pin at the jackshaft and a 10 by 9-in. pin at the driver. See Fig. 12.

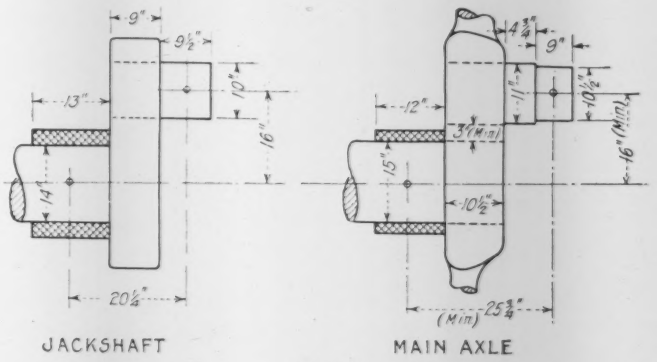

FIG. 12

It was to be noted that the pin-bearing value was somewhat high and therefore a larger pin might be necessary in a final design.

For the jackshaft and axle using possible peak loadings for strength proportions, and rod thrust at 90 deg. crank position for mean bearing loadings with due consideration for peak bearing loads, then (using a 14-in. jackshaft and 15-in. axle) the peak stress would be,

Jackshaft: $\frac{270{,}000 \times 20.25 \times 32}{\pi \times 14^3} = 20{,}300$ lb. per sq. in.

Main Axle: $\frac{270{,}000 \times 25.75 \times 32}{\pi \times 15^3} = 21{,}000$ lb. per sq. in.

Main Pin: $\frac{270{,}000 \times 9.75 \times 32}{\pi \times 11^3} = 19{,}000$ lb. per sq. in.

The bearing values would be:

Jackshaft: $\frac{134{,}000}{13 \times 14} = 740$ lb. per sq. in.

Pin: $\frac{134{,}000}{95} = 1410$ lb. per sq. in.

In a corresponding reciprocating-locomotive design the maximum main-rod load for the same weight, assuming a 30-in. stroke and a 72-in. driver, was

$$P = \frac{0.33 \times 180{,}000 \times 36}{1.41 \times 0.85 \times 15} = 120{,}000 \text{ lb. (max.)}$$

This permitted less overhang, smaller axles, pins, etc. and was but 63 to 45 per cent of the loading in a turbine locomotive.

It was interesting to note that in reciprocating locomotives, in no case were there found more than two pairs of drivers driven by one side rod, the front and rear coupling drivers in a Santa Fe type being driven by the middle or main driver.

With the larger torque loadings required at the jackshaft, the design of the reduction gearing became complicated by the very limited space available, necessitating possible overhanging of the turbines. Moreover, the rod loadings previously stated were based on the assumption that sufficient torsional elasticity was obtained at the jackshaft. Some form of torsional spring system at the jackshaft was of primary importance, to reduce possible impact loadings

on the rods and journal bearings. Finally, for sufficient gear reduction the driver diameter might be increased by the main-gear diameter for proper clearances. With the larger driver diameter and the need of limiting the throw on account of centrifugal forces and the balance, especially on the jackshaft, the rod loadings might become excessive.

Thus it was clearly seen that a direct mechanical drive was very much limited in horsepower and adhesive-force capacity, especially for the large locomotives used on American railways, unless two sets of turbines, jackshafts, etc. were used. A combined system by the use of two units, one on the locomotive driven by a high-pressure reciprocating piston and the other under the tender driven by a low-pressure turbine, had certain merits. By this method, however, considerable throttling of high-pressure steam to the low-pressure turbine would be needed for the starting period, since the discharge steam from the reciprocating locomotive during this period would be insufficient to give full-load torque. By giving greater adhesion weight to the reciprocating unit a small reduction in the initial steam consumption might be effected.

The turbine locomotive was, so to speak, wrapped up in the condenser, and the justification of the turbine was of course based on the vacuum produced by the condenser. Simple calculations would quickly show that even with poor vacuums, a reciprocating compound engine developing 3000 hp. with our gage limitations, and even with articulated running gear was constructively impossible; hence the inevitable necessity of the turbine in conjunction with the condenser. Therefore, a major problem for development included the design of a minimum-weight condenser giving a high vacuum under ordinary operating conditions and at reasonable cost.

In judging the ensemble design, a serious factor was the weight efficiency. This should not exceed 200 lb. per hp. and preferably should be around from 150 to 170 lb. per hp.

Another consideration was the arrangement of wheelbase, etc. for proper tracking and reducing concentrated axle loading without undue overall length.

A. F. Johnson[1] reminded the audience of an experiment made on a steamship some years where the designers combined a low-pressure turbine on a central screw with two reciprocating engines on port and starboard. Theoretically economical, the arrangement did not work out well in practice. He was afraid that since the turbine was a high-speed machine and was going to be coupled to a low-speed machine in a space which was decidedly limited, under conditions which made it very difficult to condense the exhaust steam, the problem was going to be very complicated.

M. B. Richardson[2] inquired whether a turbo-driven locomotive would get under way just as an ordinary locomotive could. He was also anxious to know whether a turbo-locomotive would provide the necessary back-up power when it was needed.

Prof. W. F. M. Goss[3] complimented Professor Zoelly on his energy in developing a design along such revolutionary lines. American designers were not revolutionary but had nevertheless in thirty years trebled the power and halved the fuel consumption for a given power of their locomotives. They were likely to continue with their gradual methods of development.

Charles E. Chambers[4] asked whether the Zoelly locomotive could be used for both tonnage and speed, and also whether the turbo-generator could be cut out at times, or whether it had to remain in constant use all the time.

Spencer Miller[5] discussed locomotive problems in taking timber down the mountain sides of the Pacific Coast and commented on the peculiar fact that wood was not chopped up into fuel, although excellent for this purpose, because labor costs made it a cheaper proposition to transport oil to the scene of action. He thought the locomotive problem was a very serious one on American railways, particularly in western states where serious grades had to be met. Electrification did not seem to answer as well as expected, and the power plant, he thought, would be one that would be carried on board the locomotive.

F. D. Herbert,[1] in reference to combination turbine and reciprocating engines on board ship said it was the S. S. *Olympic* which had this combination of one central turbine and two reciprocating engines, and that the arrangement had worked out satisfactorily.

L. B. Jones[2] thought the outstanding advantages of the turbo-locomotive were fuel saving and absence of reciprocating parts. He said that the limit of the power transmittable through two crank-pins had practically been reached. He was disappointed that although the General Electric Co. would promise much greater power with three-cylinder locomotives, they were not offering full turbine equipment. Mr. Jones pointed out also that since the hardest pull occurred sometimes in a tunnel, in which condensing power might be reduced, condensation in a tunnel might be a serious problem.

E. N. Trump[3] pointed out that a locomotive had been built in Europe with a uniflow engine and had been successful in operation. With a uniflow engine initial condensation was eliminated. If in addition to this a good vacuum was available, a very high efficiency indeed was possible. Professor Stumpf who had invented the uniflow cycle had also invented a system of induced exhaust, which he used to induce a draft in other cylinders. A nozzle was arranged in such fashion that the exhaust of one cylinder produced 80 per cent vacuum in the other cylinders. Professor Stumpf expected thereby to get an increase of 20 per cent in the efficiency. Mr. Trump predicted that the next American locomotive would have three cylinders with induced exhaust on a similar principle.

R. C. H. Heck[4] wished to know whether the turbo-locomotive could be depended upon for high initial slow-speed torque.

A. A. Jacobson[5] said that the introduction of the turbo-locomotive meant departure from simplicity. Perhaps it would be better to go back to the fundamentals of the locomotive and see whether transformation of energy could not be as readily effected without this complicated system of transmission. Would not the cost of the turbine locomotive exceed that of the reciprocating engine?

W. E. Symons[6] gave some interesting statistics. Last year the United States had paid $671,000,000 for fuel. The average coal consumed by a locomotive was more than $10,000 a year, while some went as high as $15,000 a year. Any device that would diminish fuel consumption was therefore worth considering. But the extinction predicted for the steam locomotive when electric cars were first introduced had not taken place, and it might be well to move slowly in the introduction of new features.

E. D. Dickinson[7] pointed out that the initial cost of the equipment was relatively of little importance. The steam turbine was now so simple, its parts so well proportioned and the electrical part of the equipment so practical that the turbo-locomotive was only a question of some one's putting up enough money to build one. The space occupied by a 2000-kw. turbine would not be much greater than that occupied by a 1000-hp. engine of the present type.

A communication received from the Aktiebolaget Ljungströms Angturbin, of Stockholm, Sweden, builders of the Ljungström turbo-locomotive, called attention to the fact that this engine had successfully run for over 10,000 miles in regular service on the Swedish State Railways since its first trial in October, 1921. In addition to low consumption of water its fuel economy was stated to be unusually high. Tests of both turbo and piston locomotives had recently been made, and the following comparative results had been obtained:

	Turbo-Locomotive		Piston Locomotive	
Weight of cars, tons	415	400	424	424
Weight of train, tons	555	541	551	580
Average speed, m.p.h.	40.9	41.5	48.7	48.6
Coal per 1000 ton-miles, lb.	39.1	45.5	130	82.9

Taking an average of the coal consumption for the two turbo-locomotive tests (42.3 lb.) and comparing it with the lower value for the piston locomotive (82.9 lb.) would show a saving in favor of the turbo-locomotive of (82.9–42.3)/82.9 = 49 per cent.

[1] George Washington University, Washington, D. C.
[2] *Railway Age*, New York, N. Y.
[3] Railway Car Manufacturers Association, New York, N. Y. Past-Pres. A.S.M.E.
[4] Central Railroad of New Jersey, Jersey City, N. J. Mem. A.S.M.E.
[5] Ch. Engr., Lidgerwood Mfg. Co., New York, N. Y. Mem. A.S.M.E.

[1] Terry Steam Turbine Co., New York, N. Y. Mem. A.S.M.E.
[2] Master Mechanic, Pennsylvania Railroad Co., Harrisburg, Pa. Mem. A.S.M.E.
[3] Cons. Engr., Solvay Process Co., Syracuse, N. Y. Mem. A.S.M.E.
[4] Prof. M.E., Rutgers College, New Brunswick, N. J. Mem. A.S.M.E.
[5] New York Central R. R., New York City.
[6] Angus Sinclair Publishing Co., New York City. Mem. A.S.M.E.
[7] General Electric Co., Lynn, Mass. Mem. A.S.M.E.

Production Control

BY GEORGE D. BABCOCK,[1] PEORIA, ILL.

The conclusions arrived at by Taylor in his paper on Shop Management in 1903, together with the methods he described, were based upon the thesis that every organized effort of human endeavor can be analyzed into its fundamental elements, and that these elements can be forecast and arranged in an orderly sequence that represents the best combination to attain the desired result. The analysis and arrangement of the elements of production brought about the two great divisions of productive effort that are characteristic of modern industrial management and production control—planning and performance.

In this paper the author presents an outline of the subject of production control in manufacturing, taking up for consideration, respectively, actual output with given equipment; preplanning; the establishment of manufacturing programs; determination of lot sizes; establishment of the production schedule; operation analysis; stores systems; dispatching of work; inspection; maintenance; and forms.

TAYLOR'S great discovery in management was that scientific analysis should be substituted for rule-of-thumb and for traditional methods. As a direct corollary of Taylor's discovery, the greater part of the burden of determining the best methods of doing work was shifted from the workmen to those charged with the management of the business. The reason for this shifting of the burden lay in the fact that such determinations involved many things that were beyond the control of the working force.

Among the items that are not within control of the workmen may be mentioned the following:

a The raw material used for production, its quality and the regularity of its supply
b The equipment provided for working this raw material
c The processes through which the raw material must pass to be converted into finished product
d The sequence in which the various parts entering into the final product are sent into the factory.

Each of these items requires detailed and extensive investigation involving heavy expense and exact technical knowledge. Obviously, it would be impossible for the workers on their own initiative to conduct these, even if they had the inclination to do so.

Taylor's conclusions and methods were based upon the thesis that every organized effort of human endeavor can be analyzed into its fundamental elements, and that these elements can be forecast and arranged in an orderly sequence. This analysis and arrangement of the elements of production brought about the two great divisions of productive effort characteristic of modern industrial management—planning and performance.

The function of planning includes all of those elements that are beyond the control of the workmen, and some of those that formerly were regarded as within their sphere. It includes a decision as to the material that will be used in the processes of the factory, the exact equipment that shall be used, the method of handling this equipment, the sequence of individual operations on each part of the product, and, in the highest development of managerial science, the time that shall be taken for each operation. It also includes provision for instructing the workmen in the methods of handling the equipment so as to insure that their performance will accord with the forecast of the planning department. Upon the degree to which an industrial establishment has developed and made use of these elements of planning, depends the efficiency of production.

Classification of Manufacturing Effort

Manufacturing can be classified into the following divisions, which range in their order of efficiency from the lowest to the highest:

a One order for one piece. The piece never to be reproduced
b One order for several pieces, never to be reproduced
c Repeat orders at irregular intervals for one or a few pieces
d Repeat orders at irregular intervals for many pieces
e Repeat orders at uniform intervals for one or a few pieces
f Repeat orders at uniform intervals for many pieces
g Continuous or standing orders for the same piece.

The industry should be classified into the above classes or combination of classes, and a control plan devised in such a manner as will secure the best results for each class. A general control plan will then be devised so as to relate the classes.

Irrespective of the class or combination of classes into which the work of a manufacturing plant falls, the efficiency of its operation will depend upon two things: (1) The efficiency in selection and use of the mechanical equipment; (2) The definite preplanning of every operation and event which takes place in the progress of the work through the factory.

The Mechanical Equipment

If manufacturing efficiency be defined by the ratio:

$$\text{Efficiency} = \frac{\text{Actual Output with Given Equipment}}{\text{Maximum Possible Output with Given Equipment}}$$

it will be evident that the efficiency of the plant does not depend upon the efficiency of the equipment per se, but upon the efficiency with which that equipment is operated.

Efficient operation depends on the following factors:

a Maintenance of equipment in perfect operating condition, so that there will be no failures during working hours to delay production
b Adequate power at the machine, so that it may be utilized to its full capacity
c Proper adjustment of machines and auxiliary equipment, and, in the case of machine tools, properly formed cutting tools
d Determination of the best methods of operating the equipment, and insistence that these methods be followed
e An adequate supply of material upon which the equipment may work, so that there will be no idle time or delays due to lack of work for any machine
f Uniformity in the quality of raw material, permitting uniform operation at a predetermined rate. For example, excessive hardness in castings will compel slower operation of machine tools, and so decrease the predetermined production.

It follows from the foregoing that several departments are concerned in the utilization of equipment to its full capacity. Maintenance, power, and adjustment of machinery are within the province of a mechanical department. Determination of methods and an adequate supply of material at each machine are functions of preplanning. Uniform quality of material is a result of careful purchasing and rigid inspection of purchased material to insure conformity with specifications.

Maintenance can be insured by the establishment of a definite routine of inspection to discover defects, wear, or misadjustment which will tend to cause failure or decrease the rate of production.

Preplanning

Preplanning comprises:

a The establishment of a definite manufacturing program
b The purchase of materials and the insurance of their delivery in ample time to carry out this program
c The determination of the methods to be used in carrying out the program
d The sequence of operations to be performed on each component part of the final product (Routing)
e The establishment of definite schedules to fix the time at which each operation in the routing shall take place
f The despatching of the work in the factory in accordance with the schedule.

[1] Mfg. Executive, Holt Mfg. Co., Peoria, Ill.

Contributed by the Management and Machine Shop Practice Divisions and presented at the Annual Meeting, New York, December 1 to 4, 1924, of The American Society of Mechanical Engineers. Slightly abridged.

ESTABLISHMENT OF MANUFACTURING PROGRAMS

Manufacturing may be divided into two classes as follows:

a Manufacturing for stock. Examples of this class are automobiles, food products, certain classes of electrical equipment, typewriters, etc.

b Manufacturing on orders to definite specifications furnished by the customer. Examples are hoisting and conveying machinery, locomotives, certain classes of textiles, etc.

A manufacturing plant may lie in either or both of these classes.

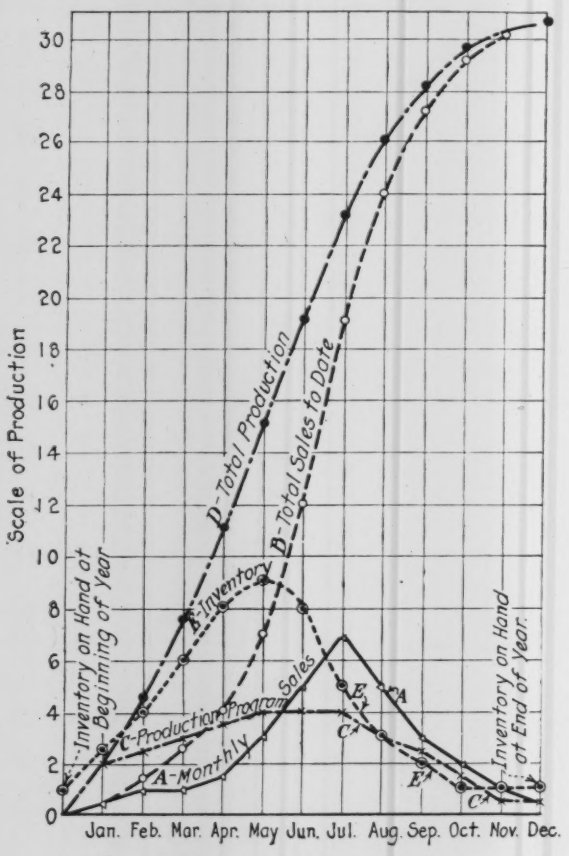

FIG. 1 DEVELOPMENT OF PRODUCTION PROGRAM.

The establishment of a manufacturing program is a radically different process in the two cases, and in the case of the plant manufacturing both for stock and to order the establishment of the program depends upon which class of work predominates.

MANUFACTURING FOR STOCK

The establishment of a program of manufacturing for stock depends first upon the probable demand for the product, whose analysis is a function of the sales department. The accuracy with which this department can forecast this demand is a measure of its efficiency.

The analysis of probable sales should show for the purpose of establishing a manufacturing program:

a The total demand for the product during a given period of time. This period should be as long as possible, and preferably not less than six months

b Whether or not this demand will be at a uniform rate. If not uniform it should show:

c The variation in demand from month to month, or even for shorter periods if possible.

With these data before it, the management may then plan the productive activities of the plant so as to meet this demand in the most economical manner. Fig. 1 is a graphical presentation of a sales analysis and the resulting production program. Curve *A* represents the expected sales month by month. Curve *B* shows these monthly sales accumulated to give the total sales at any time after the beginning of the sales program. Curve *C* shows the production program month by month, and Curve *D* the monthly production accumulated to show the total production at any time during the program. Curve *E* is the inventory. The production program should be so laid out that the inventory will never fall to zero. Another curve might be added to show the actual total production in comparison with the planned production. If the production management, however, is efficient, this curve would fall exactly upon Curve *D*.

It will be noted that the monthly production program does not follow the monthly sales program. The reason for this in the case at hand is that there is always a minimum size of lot that it is economical to put through the plant, and this minimum may be larger than the expected sales at the beginning and end of the program. Furthermore the expected sales at the middle of the program are greater than the manufacturing facilities of the plant and provision must be made for these by building up the inventory during the early part of the program.

The establishing of uniform manufacture with a minimum amount of labor turnover, and at the least cost, is largely dependent, in an industry where sales vary from month to month, upon the starting of the manufacturing program for a given sales program at the proper time. This proper time is such that sufficient units of product will be built ahead of the sales peak, so that the sales peak will reduce the accumulated product to a normal or predetermined inventory. There may be cases where this slow increase of inventory must be started several months in advance of the peak, for the further away from the peak the accumulation can begin, the less will be the disturbance created by the peak, and the maximum possible operating efficiency will result. A graphical analysis of this character will give a clearer idea of the manufacturing problem than any other presentation.

The rate of production having been decided upon, the next decision to be made is as to the size of the various lots of product that must be put through the plant, and the intervals at which these lots must be started in order to meet the production program. These decisions having been made, the production follows the routine described later.

The production program should be analyzed with reference to economical utilization of personnel, as well as for production according to a sales program. If the final product is composed partly of parts made in the factory and partly of outside purchases, the schedule of deliveries of the outside purchased parts may be made to conform to the sales program, while the production of manufactured parts is maintained at a uniform or nearly uniform rate.

Fig. 2 shows such a schedule as laid out for automobile manufacture. The sales program shows a variation of 70 per cent. The schedule of assembly adopted to meet this variation shows a variation of 30 per cent. The schedule of production of manufactured parts is uniform and has zero variation. Analysis of the product developed the fact that 60 per cent of the cost of materials entering the final product was contained in 59 out of 1330 parts. Of these 59 parts, 44 parts, representing one-half of the cost of material, were purchased finished parts which did not enter into the production schedule until the final assembly. The balance of the other parts entering the car represented such a small relative material value that it was possible to operate the productive labor manufacturing parts on a uniform schedule. The final assemblies, however, were made according to a variable schedule as shown in Fig. 2.

In arranging the variations of the assembly schedule, a variation in labor of not more than 30 per cent was permitted; i.e., not over one man in every three on the assembly force was allowed to be changed, two-thirds of the force thus consisting constantly of trained men. The cars were assembled according to the schedule shown in Fig. 2, and the high-cost materials noted above were ordered for delivery to agree with this schedule. The capital invested in inventory was thus kept at minimum, both for purchased and manufactured parts.

DETERMINATION OF LOT SIZES

Several factors enter into the determination of the most economical size of lot to be used in manufacture. The first and most important of these factors is previous experience with a similar product.

The following specific considerations must be taken into account:

Small lots are indicated by:

a High cost of material
b Large unit bulk of material
c Long time required for various operations on a part
d Probability of change of design
e Reduction in inventory
f Conservation of floor space
g High unit weight of material
h Use of perishable materials in processes
i Early entry of part into sub-assembly

Large lots are indicated by:

a High average cost of machine set-ups
b High despatching, inspection, and trucking charges
c Rapid production, even with elaborate set-up
d Reduction of spoilage
e In general, all preparation charges should be satisfied with largest possible lots.

A number of formulas have been prepared at one time or another for the mathematical determination of lot sizes. The more accurate of these are based upon material cost per piece, time

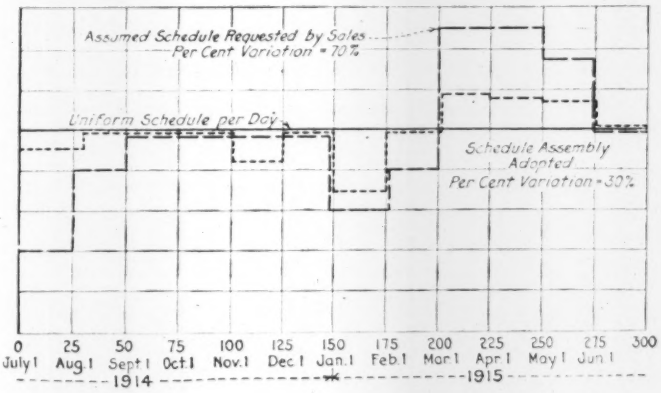
Assumed Schedule Requested by Sales
Per Cent Variation = 70%
Uniform Schedule per Day
Schedule Assembly Adopted
Per Cent Variation = 30%
0 25 50 75 100 125 150 175 200 225 250 275 300
July 1 Aug. 1 Sept. 1 Oct. 1 Nov. 1 Dec. 1 Jan. 1 Feb. 1 Mar. 1 Apr. 1 May 1 Jun. 1
1914
1915

FIG. 2 AN AUTOMOBILE MANUFACTURING SCHEDULE

required for operations, interest charges on material in process, rental charges for space occupied, wage rates, etc. Accurate formulas taking all these facts into consideration lead to cubic equations which are difficult of solution. The author has found it preferable to determine lot sizes by trial and error rather than by applying a formula.

MANUFACTURING TO ORDER

A plant manufacturing to order obviously cannot plan its production in advance of the receipt of orders. Preplanning in this case involves only the analysis of the order to determine the material that must be purchased and the time required to procure it, and to determine the operations that are necessary, the sequence in which they shall be performed, and the time at which they shall start in order to meet the delivery date specified in the order. These determinations must be made with reference to orders already in process, and the new order must be fitted into the manufacturing schedule in such a manner as not to create disturbance. The purchasing is often simplified by the fact that most manufacturing concerns of the class under discussion operate in a somewhat limited field as regards product, and the raw material used for all orders is of the same character. Thus a company specializing in hoisting and conveying machinery will use for practically all of its orders pig iron, sheet and bar steel, alloys, and certain standard manufactured products such as rivets, bolts, pillow blocks, chain belts, etc. These it will keep on hand in sufficiently large quantities to fill its requirements over a certain period, purchasing in large lots whenever the stock falls to a predetermined minimum. If the analysis of the order shows that there is sufficient raw material available to fill it, production can begin at the earliest date on which machine capacity is available.

Another class of manufacturing which is to order, but which has many of the characteristics of manufacturing to stock, is found in certain phases of the textile industry. In the manufacture of woolen fabrics for dress goods, orders are taken by means of samples for varying quantities of product of different grades and patterns from numerous customers. The small orders for each fabric are combined into single manufacturing orders, from which a manufacturing program is built up. Since the samples are sent out and orders taken before the manufacturing season starts, the mill is enabled to plan a production program in its entirety before actual production begins.

Where manufacturing for stock predominates, the stock program is laid out as heretofore described, and the product made on order is fitted into the stock program. This may be accomplished by ascertaining what proportion of the machine capacity is not utilized over a given period for stock manufacture, and then scheduling the special product through this excess capacity as if the balance of the equipment did not exist.

Where the stock production is of minor importance, it is scheduled on the basis of the excess machine capacity available. In doing this, if the stock program extends beyond the time for which special orders on hand will utilize the equipment, care must be taken to reserve sufficient capacity to handle future special orders. The capacity so reserved should, in general, be equivalent to the capacity required for special work during some definite preceding period, say, from one to six months.

THE PURCHASE AND STORAGE OF MATERIALS

In purchasing material these considerations must be taken into account:

a Quality
b Price
c Time of delivery.

Quality is important, especially where the final product must meet specifications as regards strength, durability, etc. It is also important in that it may affect the time required for completing the processes in manufacture, and if of different quality than that upon which the manufacturing program is based may require much more time at each operation.

All things being equal, the lowest-priced material is to be desired. If, however, the quality is lower, or delivery cannot be insured at the desired time, price becomes less important than other considerations. The time required for delivery is extremely important in laying out a manufacturing program. Since production cannot begin until there is material upon which to work, ample time must be allowed in which to procure material. It is advisable to tabulate the maximum and minimum number of days required for each item of material that is commonly bought, as a guide in placing orders. The reputation of the seller for keeping delivery promises should also be considered, and preference given to one who can be relied upon, rather than to one who will promise a prompt delivery but whose promises are unreliable. It is better to accept a later delivery date, with the knowledge that it will be met, than an earlier date where there is reason to doubt that delivery will be made on that date. Another factor to be considered is whether the material shall be bought from the mill or from a warehouse or jobber. Goods obtained from the mill usually cost less, but the time required for delivery is longer.

It should be borne in mind that the shortest time required for the execution of a manufacturing program is the longest time required to procure any item of material plus the time required for the longest process through which that material must pass. The importance of accurate forecasts of the time necessary for purchasing is thus evident.

In the routine of establishing a manufacturing program, then, the first item is to ascertain when the first orders for material must be placed. To do this it is necessary to work back through all the processes of production, starting from the time when the finished product must begin to leave the factory. The methods of doing this are explained later. This analysis of the operations fixes the time at which the material for each process must be available. To this time is added the length of time necessary to obtain prices and secure delivery. Having thus fixed the number of days that must elapse between placing the order for material and the completion of all the manufacturing operations, exact dates can be set for the beginning of each event in the production program. This is known as "scheduling."

Establishment of the Production Schedule

In making a schedule for manufacturing, the completion of the final operation is the starting point. The number of days that must be allowed for each standard lot or operation must be ascertained. The length of time that must be allowed between operations for movement of material, for inspection, for seasoning or for any other purpose whatever also must be determined. These various intervals are then to be added progressively to show the number of working days prior to the date set for completion of the first lot of the product that each operation should begin. Then by means of a calendar, from which Sundays and holidays are omitted, the actual calendar dates for these operations can be fixed when the date of completion is known. Table 1 is an example of the fixing of the calendar dates for the principal events in the production of a cylinder for the gasoline motor of a tractor, deliveries of which are to begin on July 1.

TABLE 1 PRODUCTION SCHEDULE FOR CYLINDER OF TRACTOR MOTOR

	Finished tractor	Assembly	Machine shop	Rough stores	Purchase	Engineering
No. of days in department.	0	22	24	17	60	10
Working days ahead of completion	0	22	46	63	123	133
Date work should be finished	July 1	June 5	May 7	Apr. 7*	Feb. 7**	Jan. 25

* Date on which invoice should be paid.
** Date on which purchase order should be placed.

This cylinder is a machine-shop product, on which many operations are performed. The actual working schedule would show in detail every operation, and the date at which that operation should begin. Several ways of doing this are in common use, and each one has its advantages. No one method can be said to be the best, for the reason that the method to be adopted should depend on the circumstances surrounding its use. Two methods that have proved very successful may be cited.

The first uses what are known as "route sheets." One route sheet is used for each item of product, and lists every operation in its order through which the item is to pass, together with the machine, and the date on which the operation is to be performed. Progress of the work is noted by checking off each operation as it is completed. When an operation is checked off, the orders for the next operation are issued on the date set opposite that operation. The route sheet may be usedfor large quantities or for single piece orders.

The second method is semi-graphical. Consider for a moment a single item of product that is made up of many parts. If a horizontal line is divided according to scale of working days, it can be made of such a length that the number of days represented by it is the total time from the completion of the design to the delivery of the final product. Then the schedule can be laid out by noting on this line, according to the scale of working days, the number of days prior to completion that each operation shall begin. The time required for this operation may be shown by blocking off on the line a distance equal in length to the space representing the number of days required. This space is laid down from the point denoting the beginning of the operation toward the point representing the completion of the product, known as the "zero point." Another scale, also graduated to the same scale of working days, but reading from right to left in calendar dates, with Sundays and holidays omitted, is then laid along this line of operations, with the date set for completion placed on the zero point. The calendar date on which any operation should begin may be read directly on this scale. If for each item that enters into the final product a similar horizontal line of operations be laid out, there is presented a graphic picture of the production schedule for the entire product.

Such a graphical schedule may be used to control production by the addition of another scale, equal in length to the total time shown on the schedule for the part requiring the longest time. This scale is graduated to read in items of finished product, and is numbered from right to left, from zero to the total number of items in the program. This scale is movable, and at the beginning of the program is set with its zero at the extreme left of the schedule, and corresponding to the first operation shown. It is moved to the right each day a distance equal to one day on the horizontal scale. Whenever the zero of this moving scale passes over any operation posted on the schedule, it is an indication that that operation should begin. The necessary orders should then be issued for such operation.

Progress of the production program can be recorded by writing over each operation in terms of units of final completed product the quantity of product that has passed through it at the end of any period of time. If the number so written corresponds to the number immediately over it in the moving scale or schedule tape, the operation in question is in accord with the schedule. If the number is greater than that on the schedule tape, the operation is ahead of schedule. If it is less, the operation is behind schedule. The number of days that it is off schedule is ascertained by locating on the schedule tape the number corresponding to that written on the operation, and noting on the scale of work days the interval between the position of this number and the operation in question. (See Fig. 3.)

This method is the basis of control boards that have been used to control operation in repetitive work, where the product is put through in lots, as in the automobile industry. If it is desired at any time to increase the schedule, this may be done by decreasing the time interval at which lots are started in the factory.

The route sheet and the control board are adapted to industries where the product is made up either of single pieces or of a large number of parts that are assembled into a final product. Where the final product is not an assembly, such as cloth fabrics, or where it is an assembled product made in such large volume that the manufacture of each part is on a progressive or continuous basis, scheduling is determined by the speed of the final operation. This determines the amount of product that can be produced per day, and the purchase of material is arranged so that there will be daily supplied to the factory equipment an amount equal to the daily production. The only date to which much consideration must be given is the date on which the first purchase order must be placed. This is fixed by the time required for process up to the final operation plus the time required to procure material. After the flow has once been established, the operations become automatic in their relation to one another. In certain continuous processes the speed of production is governed by the speed of conveyors which transport the work in process from one operation to the next. If it is desired to speed up the schedule in such a case, it is done by increasing the speed of the conveyors, or by decreasing the intervals at which material is delivered to them for the first operation. Additional machine capacity must then be added at each operation to take care of the heavier flow of work.

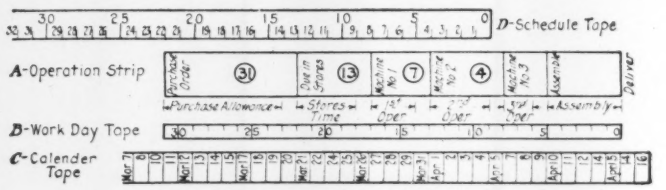

Fig. 3 The Principle of Graphic Control

Operation Analysis

It is evident from the discussion on scheduling that an important feature is exact knowledge of time required for each operation. It is not enough that the time for an operation by the method in common use be known. For the most efficient management, the time of the best method should be found. This includes both the best method of performing each individual operation and the best sequence of operations.

A prerequisite to efficient scheduling, then, is an exhaustive analysis of each operation to discover the best method and equipment for performing that operation. This also should include an investigation to discover and eliminate all causes that will hamper production, and to standardize the conditions under which production shall be carried on.

For instance, in machine-shop work there may be several methods available for performing a single operation. Boring and turning may be done, for example, in a lathe, a boring mill, or a turret lathe. Considerations that govern the choice of the particular machine in this case might include the size of the work, the num-

ber of pieces to be made, the quality of finish demanded, the relative time by each method, the number and cost of special tools, and the time required to set up each machine for the operation in question. The analysis should determine, first, the method, and next standardize the conditions of using that method. In the machine shop the shape of the cutting tools has a marked effect on production. It is highly important, therefore, that these be standardized and that the same shape and method of heat treatment be used for the same work every time that it occurs. The best combination of speeds and feeds should also be ascertained for each job, and the workmen should be required to use these combinations. The machines themselves should be standardized so that identical work can be done in machines of the same class. If this is not done, there may be a congestion of work at some machines while others of the same class are idle because they are less efficient than the others. Or if the work is routed to these inefficient machines it may not be possible to do it in the time allowed in the schedule, which should be based on the best method. The schedule may thus be upset at these machines with disastrous results so far as controlled production is concerned. Standardization is a great aid to simplification of routing and scheduling.

QUALITY OF MATERIAL

Uniformity in quality of material is an important factor in scheduling. In the analysis of a job a certain standard of quality is assumed, and all time estimates are based on this. Variations from this quality may seriously affect the time of operation. Uniformity in quality can be insured as a rule by rigid inspection of raw material accompanied by tests to determine quality. All material not up to the standard set should be rejected.

DETERMINATION OF OPERATION TIME

The time required for an operation can be ascertained in a number of ways. Time study is one. Estimation, based on similar work or on previous performance, is another. In work where the machine has a definite output per unit of time, as in a spinning frame or loom in textile work, the time is a matter of calculation based on the quantity of material to be handled and the speed of the machine, with a certain percentage added to cover unavoidable machine delays and the setting up of the machine. In machine-shop work, where the time required for handling the work and for setting up the machine is relatively small compared to the time actually required for cutting metal, calculation based on feed and speed and the dimensions of the work, plus a percentage of flat time allowance for handling, etc. may be sufficiently accurate.

While it is desirable, of course, that the time allowed for an operation be as accurate as is possible, it is unwise to strive for absolute perfection when the advantage to be gained is small and the cost of the last few per cent in the perfection scale is high. Thus, if the handling time on an operation is two minutes, and the machine time thirty minutes, an error of even 25 per cent in estimating the handling time will make an error of only about one per cent in the total time of the operation. The actual length of the job, however, must be considered in deciding whether or not the error due to estimating will be serious. In the case just considered an error of 25 per cent or one-half minute per job would amount to but seven minutes per day, since less than sixteen jobs are all that are possible in an eight-hour day, and the maximum difference in production would be one piece per week. If, however, a job requires but three minutes with a handling time of one minute, an error of 25 per cent in estimating handling time would be more serious. The difference in product would be one piece per hour, or eight per day.

Time study has not the same relative importance that it held at the beginning of the movement for better industrial management. Several reasons for this exist. In the first place, an enormous amount of time-study data has been accumulated and published. These data cover the smallest elements of the work to which they relate, and by combining these elements the time for the great majority of operations can be predetermined with considerable precision. Another reason is the standardization of equipment. Formerly hardly two machines designed for the same work were alike in their characteristics. At the present time there is a remarkable similarity in the characteristics and capacities of machines of the same class, even when built by different makers. Then, too, improvements have been made so that the time for adjustments and manipulation is much reduced and as a rule forms but a small fraction of the total time for operation. Machine capacity for a given job can be determined from tabular data or by calculation, aided by slide rules if desired, and in most cases the machine time is the most important factor. There are many things to be done in the line of standardization, establishment of material control, etc. before a factory is ready for time study.

STORES SYSTEM

The stores system includes the storage of material and the accounting for it. The accounting system should be so designed as to show at all times the quantity of material of each class and size actually on hand in the storeroom, the quantity on order but not yet delivered, the quantities apportioned to or reserved for various orders not yet in process, and the particular orders to which these quantities are assigned, and the balance that is available for future orders. The stores-ledger sheets should also show the location in the storeroom of every item carried, and the unit value of these items. This last feature enables inventory to be taken directly from the stores ledgers without the necessity of a physical inventory in the storeroom.

In the storeroom, each commodity should carry a bin tag, on which is entered the quantity of material in storage. Additions to the quantity on hand should be entered on the tag, and added to the total already entered. Material issued should be deducted from the bin-tag total and a new total brought down. The total quantity shown on the bin tag should always correspond with the quantity shown as on hand in the ledgers, and frequent comparisons should be made to detect errors.

No material should be permitted to leave the storeroom, except on the authority of a stores issue order, signed by some one with the necessary power. Neither should material be received unless accompanied by the necessary documents to show its origin. These, as well as the stores-issue orders, should pass through the hands of the stores-ledger clerks to insure that the ledgers will be kept up to date.

DESPATCHING OF WORK

Except in continuous or progressive manufacturing, despatching is essential to make preplanning effective. Despatching is the assignment of jobs to machines or workmen in the sequence and at the times determined by the schedule established in advance. Under modern management, the foreman or overseer has no authority to dictate when or where any operation shall be performed. The time of its performance is fixed by the schedule, and the place by the job analysis. Work orders are then prepared for each operation shown on the schedule and filed according to the schedule dates. Each day the orders for that day are issued to the workmen concerned. If the work called for by any work order is not available, due to previous operations being incomplete, the order is marked in some manner to make it conspicuous, and pressure is exerted until the work is once more on schedule. Despatching thus may be performed by a well-trained clerical force and foreman relieved of routine. By noting on them the time of issue of work orders, and the time of completion of work, the despatching may be used as the time-keeping system of the shop and also used to accumulate data for the cost accounting.

INSPECTION

Defective product results only from the two causes of defective material and poor workmanship. The inspection function can be arranged to prevent both of these causes. Inspection of incoming material has already been discussed. As a preventive of poor workmanship, inspection should take place at the beginning of a job. The inspector should ascertain whether the product at each operation is being made according to the specifications, and if not, point out the cause of variation. He should not leave until the workman has produced and is producing perfect work. The despatching of work should include the directing of the inspector to operations that are just being started.

This inspection of work in process should not operate to dispense with final inspection of each part of the product after the last

operation. In final inspection, the inspector should be advised as to just what faults or variations from standard are to be looked for, and the permissible amount of variation. The provision of a definite routine for inspection will render it more efficient and less costly.

Maintenance

The duties of the maintenance department include the repair of equipment that fails or wears in service, and the establishment of a system of inspection and preventive repair that will remove causes of failure or make adjustments for wear before failure occurs. Failure of equipment in service has many disadvantages. Not only is there the cost of repair, but also the wages of idle workmen while repairs are being made, and the loss of profitable production.

The maintenance department can avoid all or nearly all of these difficulties by establishing a routine that will insure inspection of the equipment of the plant at regular intervals.

Forms

To carry out the routine of any system of management, certain forms are necessary. The system of forms should not be confused with the system of management. The latter is an organized scheme for obtaining certain results, based on definite principles. The former is simply the mechanism provided for the latter. And let it be stated here that scientific management is not a particular system nor a set of forms. These are but a means to an end. They represent only the easiest and most economical method of applying the principles of management to the case in hand. Two plants may have entirely different systems and their forms may be totally unlike, and still both plants may be truly representative examples of scientific management. Forms are merely a permanent statement of the wishes of the management in regard to the matters to which they relate. They replace orders and instructions that in the early stages of an enterprise or program are given by written memoranda or verbally. When an order or instruction is repeated sufficiently often to warrant it, a form is devised to convey the same information, concisely and definitely, leaving only the necessity of writing in the figures, dates, and symbols to make the form complete. Forms may also be considered as a delegation of authority to the individual authorized to issue them, the authority, however, extending only to the subjects covered by the form.

Discussion

ROBERT T. KENT[1] agreed with the emphasis placed on preplanning as the keynote to all industrial management, necessary not only in production but in purchasing and finance. As an answer to the critics of scientific management, he pointed out that the Holt Manufacturing Company had recently put through on schedule a program involving upward of 8,000,000 operations. Scientific management did work, apparently.

Fred. J. Miller[2] did not think that the control board mentioned by the author had the same value or efficiency as the system of charts and records devised by H. L. Gantt.

Ralph E. Flanders[3] disagreed with the author on one point. He thought that a complicated mechanism, with a great many parts and sub-assemblies, passing through the shop at a relatively high rate of production, was nevertheless susceptible of a manufacturing scheme and a production control as standardized as in the continuous production of the Ford automobile.

J. P. Jordan[4] suggested that it would be of distinct advantage to the movement for promoting better management of manufacturing institutions to make it very clear always that highly detailed methods of production control applied only to plants where repeat or continuous work was going on. Highly detailed methods, absolutely indispensable in some industries, were frequently applied in operations calling for less control with disastrous results. There should be careful study as to the extent of control advisable in any particular case.

[1] Supt. of Prison Industries, Albany, N. Y. Mem. A.S.M.E.

[2] Member, Public Service Commission of Commonwealth of Pennsylvania, Harrisburg, Pa. Past-President, A.S.M.E.

[3] Mgr., Jones & Lamson Machine Co., Springfield, Vt. Mem. A.S.M.E.

[4] Cons. Indus. Engr., Stevenson Corpn., New York, N. Y. Mem. A.S.M.E.

Howard Coonley[1] agreed that clever formulas had been developed for determining the economical lot size, but thought that they were usually too complicated for practical use. He did not think that estimates of the sales department should be used as the sole basis of a production program. These estimates might be supplemented by the work of planning and statistical units that would weigh sales estimates against independent forecasts, based on sound economic data. In regard to the author's remarks on the subject of maintenance, Mr. Coonley said that many failures of equipment could be prevented by regular inspections, carefully planned and scheduled.

The author in closing the discussion met criticisms of the control boards by citing the experience of the Holt Manufacturing Co. and received acknowledgment from the speaker that he did not know any of the details or performances of this mechanism. Regarding the matter of machine-hour cost, a fundamental in the application of the principles described in the paper was that a part should not be produced that could be bought for less money outside of the plant. The machine-hour cost method alone could determine whether production in the plant or purchase outside was the cheaper.

Elements in Lowering Machine-Production Cost

WHEN improvements are made in existing machinery to increase production, or the same result is effected through the substitution of newer machinery, or the addition of equipment, then it is the fair thing to change rates and bonuses for production to at least the extent to which the change is responsible, if extra effort of the operator is not created thereby.

Machinery production cost, cannot however, be considered merely as to the single machine, for complete production is almost invariably a matter of group machinery operation. The cost, therefore, of this production, if considered from the economic point of view, as of course it must be, involves two important elements. The first of these is straight-line and continuous production. If the best value is to be obtained from one machine, it is important that the raw or semi-finished material be promptly fed to it and taken away from it. When, therefore, the grouping of machines to bring about finished production is considered, both the questions of readiness of material and transportation become important. A chart of the movement of material in many industrial units frequently presents a picture of tangled threads, involving considerable extra labor cost, whereas as nearly as possible, a single line should represent this movement of material.

A second most important element involves the equipment of the industrial unit both as to the machines themselves and transportation to, between and from them. A proper scientific-management theory involves the utilization of the most skilled labor to the maximum, aiding it in less skilled work by less skilled labor. A second scientific-management theory involves the most economic results with the most helpful equipment. Transportation should be arranged not merely to make it as prompt as possible but most economical and independent of labor cost. Perhaps an outstanding example of such a solution is the Ford method of assembling. Here again a proper cost system may quickly operate with data and calculation to develop a formula, and to show savings to be made by installation of equipment, both to facilitate production and cheapen transportation. When this is done, a proper balance between cost of equipment and its liquidation by savings may be made with consequent decision.

These things are, of course, reasonably well known as to their theory, but their practice is too often left to chance inspiration and suggestion instead of careful and competent survey. There is in mind one case in a well-managed company, where only recently, after a number of years' operation, it was discovered that the introduction of simple, quick-handling equipment cut the operation cost in half. There are many such conditions which would develop from a practical understanding or comprehension that, in these days of shorter hours and increasing labor cost, the great need is for labor-saving machinery and equipment. (*Management and Administration*, February, 1925, pp. 150–151.)

[1] Pres., Walworth Mfg. Co., Boston, Mass. Assoc-Mem. A.S.M.E.

Test of a Prosser-Type Reciprocating Steam Engine

By L. V. LUDY,[1] LAFAYETTE, IND.

This paper contains a description of the Prosser reciprocating steam engine and the results of 53 trials made of it at Purdue University. The distinctive feature of the engine is the steam jacketing; not only the cylinder walls but the heads, piston, piston rod, and valve case are steam-jacketed. The trials were made at approximately 133 lb. per sq. in. gage pressure with saturated steam and with steam superheated 100 deg. fahr., and with back pressures of 7 and 3 lb. per sq. in. gage, atmosphere, and 21 in. vacuum. It was found that the efficiency ratios varied from 0.626 with saturated steam and 21 in. vacuum to 0.774 with superheated steam at atmospheric pressure.

THE reciprocating steam engine as it is known today was developed by James Watt in 1782, and it is rather remarkable that so many sound thermodynamic principles were embodied in his patents. For many years it was the chief source of power for manufacturing purposes and the generation of electrical energy. The steam turbine because of its adaptibility to large units has superseded the steam engine for electrical-power-station work, but there still remains an important field for efficient reciprocating steam engines of varying capacities as high as 500 hp.

Ever since the advent of the steam engine, engineers have endeavored to improve its economy by devising ways and means of reducing losses in the cylinder. As is generally known, the greatest loss occurs due to the relatively large amount of steam which is condensed in the cylinder before it has an opportunity to expand and do work on the piston. This loss is due to the alternate heating and cooling of the cylinder walls, piston, etc., made necessary by the cycle employed. In modern steam engines the heating and cooling effect is much less than formerly, but the loss due to this cause is still very great. In studying this phase of cylinder losses in steam engines, J. G. Prosser developed ideas which were incorporated in the engine upon which this test was conducted.

Most engineers are familiar in a general way with the thermodynamic losses which occur in the cylinder of a reciprocating steam engine. For this reason it seems unnecessary to refer in detail to them. It is perhaps sufficient to state that in the practical operation of a steam engine a certain amount of heat must be rejected in the exhaust, which under a given set of conditions cannot be reduced; and that the transfer of heat to and from the steam between the initial and final temperature conditions results in making unavailable a large amount of heat for transfer into useful work. This latter loss can be reduced. Theoretically the heat change in the cylinder, between the initial and final temperature conditions should occur adiabatically; that is, in the ideal expansion no heat is lost as heat. In practice the adiabatic change cannot be secured, but by applying certain thermodynamic principles it can be made to approach the ideal condition.

The methods commonly employed in approaching the above conditions are

a The use of superheated steam
b The use of steam jackets
c The use of compound cylinders.

The first two methods are made use of in the Prosser engine. Neither of these methods is new, but the steam-jacketing feature is carried out differently, in that in addition to the cylinder, valve case, and heads being provided with steam jackets, the piston and rod are hollow and supplied with high-pressure steam for jacket purposes.

[1] Professor of Experimental Engineering, Purdue University.

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THE PROSSER ENGINE

The general appearance of the Prosser engine is quite similar to engines built by several well-known manufacturers. That used in this investigation was a single-cylinder, simple, horizontal, high-speed engine designed for direct connection to an electric generator. A general view of the engine is shown in Fig. 1. Provision is made for supplying high-pressure steam to the steam jacket surrounding the cylinder and valves, to the hollow space of each cylinder head, and to the piston and piston rod. The general arrangement of all jackets is indicated in the sketch presented as Fig. 2. The manner in which steam is admitted to the hollow portion of the piston and piston rod, through the piston tail rod, is indicated in Figs. 1 and 2. The condensate which accumulates in the cylinder and

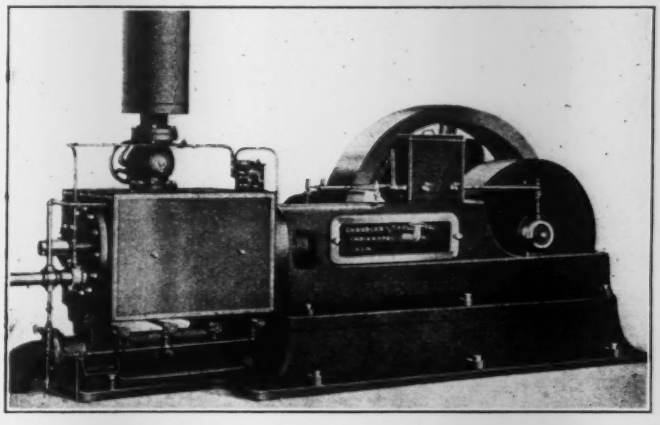

FIG. 1 GENERAL VIEW OF PROSSER STEAM ENGINE

valve-case jacket and the cylinder-head jackets is conducted by gravity through suitably arranged pipes to a trap below. The condensate from the hollow piston rod and piston collects in the lower portion of the hollow piston and is removed through a small ball check valve and drain tube by the pumping action of the parts. Small hand-operated air cocks located in the outer portion of the piston rod and at other convenient places provide a means of removing air from the steam jackets. The inner portions of the surfaces of the hollow piston and cylinder heads are grooved with small spiral grooves in order that the heating surfaces may be more effective. All surfaces of the cylinder block and heads exposed to the outside atmosphere are lagged with asbestos insulation in order to reduce radiation losses to a minimum.

The valve mechanism consists of two piston valves, one being 7 in. and the other 4 in. in diameter. The larger of the two, the main valve, is operated by an eccentric under control of the governor and regulates the admission, cut-off, and release of the steam. The smaller valve is operated by a fixed eccentric attached to the engine shaft, its function being to control the point of compression independently of the main valve. All pockets in the valve chamber which might fill with condensate and wet the surface of the valve in its movement are drained through a small ($\frac{1}{8}$-in.) pipe.

The engine is equipped with a shaft governor of the Armstrong type, which regulates the stroke of the valve to suit load conditions by swinging the eccentric across the shaft. The governor is provided with a dashpot to dampen the vibration tendencies of the governor parts. Lubrication of the piston and valves is accomplished by means of a mechanical lubricator, while the crosshead, guides, valve rod sliding-block guides, eccentric, main bearing, crank and crosshead pins, etc. are lubricated by gravity through the sight-feed oilers. The outboard bearing of the engine shaft is lubricated by ring oilers.

ARRANGEMENT OF EQUIPMENT FOR TESTS

In anticipation of the test a concrete foundation was constructed, having a floor area of 12 by 15 ft. and a depth of 8 ft. Condenser equipment, an independently fired superheater, an absorption brake of the Prony type and all necessary gages, thermometers, condensate-weighing tanks, condensate-weighing scales, etc., were provided. The engine shaft, instead of carrying the armature of an electric generator, for which it was designed, was fitted with a heavy cast-iron brake pulley.

Steam was supplied through a 4-in. branch pipe from the steam main, the exhaust being conducted to a surface condenser through a 7-in. pipe. The condenser is fitted with a circulating-

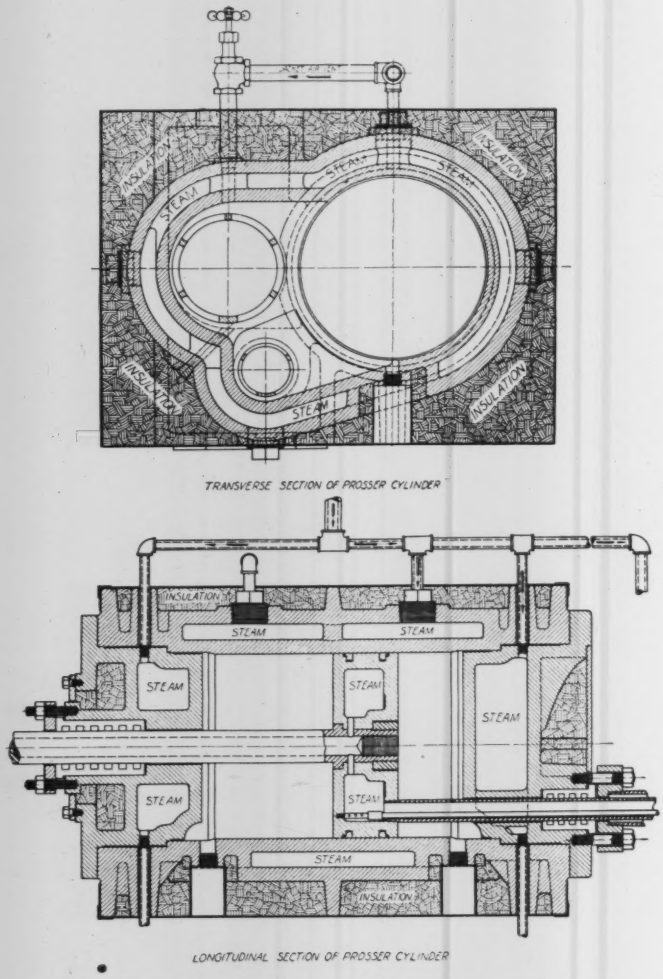

FIG. 2 SKETCH ILLUSTRATING ARRANGEMENT OF JACKETS

pump cylinder and a wet-vacuum-pump cylinder, both of which are direct-driven by a steam cylinder located between them. Circulating water was supplied from the service lines of the city water company through a 4-in. pipe. Located in the exhaust pipe above the condenser was an angle valve by the use of which any desired back pressure could be maintained. Two 1-in. globe valves connected by short nipples to the top portion of the condenser shell were opened for operation at atmospheric exhaust, and at times were opened slightly to assist in maintaining a constant vacuum.

For saturated-steam tests the steam was taken directly from the main steam line through the 4-in. branch pipe and separator mounted directly above the throttle. For superheated-steam tests valves were adjusted in the main steam line so as to pass the steam through the superheater.

The condensate from the condenser was weighed in steel tanks mounted upon platform scales. That from the different jackets was conducted from manually operated traps through small pipes to a steel tank, partially filled with cool water, mounted upon platform scales where it could be weighed, the purpose of the cool water being to prevent vapor loss. One trap was used to collect the condensate from the cylinder-block steam jacket, one for the steam jackets of the cylinder heads, and one for the hollow piston and piston rod.

A throttling calorimeter was connected to the steam supply pipe immediately above the throttle, a standard sampling tube being used. New Crosby steam-engine indicators were used in determining the indicated horsepower developed. A counter was attached to the free end of the engine shaft for determining the speed of the engine. A cast-iron exhaust manifold conducted the exhaust steam from both ends of the cylinder to the exhaust pipe. Thermometer wells were placed in each branch of the exhaust manifold to permit a temperature determination of the exhaust steam. A pipe attached at the mid-point of the exhaust manifold connected to a mercury vacuum gage for use when operating at a vacuum, and to a compound gage which indicated the back pressure in pounds per square inch or vacuum in inches. A thermometer well located on top of the cylinder casting, extending into the jacket space, was used in determining the temperature of the jacket steam. The temperatures of the steam entering the cylinder were read by a thermometer placed in a well located immediately above the throttle. The steam pressure used was indicated by a pressure gage attached at a point just above the throttle.

TABLE 1 ENGINE AND BRAKE CONSTANTS

Diameter of cylinder, in.	15
Length of stroke, in.	16
Diameter of piston rod, in.	2¾
Diameter of tail rod, in.	1¾
Diameter of steam valve, in.	7
Diameter of auxiliary exhaust valve, in.	4
Head-end clearance, per cent	4.71
Crank-end clearance, per cent	4.55
Length of brake arm, in.	63-3/32
Weight of brake arm and block, lb.	159.75
Diameter of brake wheel, in.	72
Width of face of brake wheel, in.	14

Table 1 gives essential dimensions and constants of the engine and brake.

THE TESTS

The tests were conducted in the Mechanical Laboratory, Purdue University, under the immediate direction of the author, and were made possible through the coöperation of the Chandler and Taylor Company of Indianapolis, Ind. Acknowledgment is hereby made of the able assistance rendered by R. G. Paddock, instructor in mechanical engineering, and eight senior mechanical engineering students.

The engine was operated for two months in preliminary trials, using different valve settings. Following this preliminary work a series of trials was begun, consisting of trials made at net brake loads varying from 100 to 700 lb., at a vacuum of 21 in. of mercury, at atmospheric exhaust, and at back pressures of 3 and 7 lb. per sq. in. gage. The trials were made at a nominal engine speed of 215 r.p.m., both saturated and superheated steam being used. A superheat of approximately 100 deg. fahr. and a steam pressure of approximately 133 lb. per sq. in. gage were employed.

In all trials the engine was operated under the desired conditions for a sufficient length of time to permit all temperatures to become constant before observations for record were taken. This method was followed in consecutive trials, even though the only change made was in the brake load carried. Great care was exercised in taking all readings, the Code of The American Society of Mechanical Engineers being followed as closely as possible. Observations were made at time intervals of five minutes, and as nearly as possible simultaneously. Because of the time available in which to complete the work, it became necessary to limit the duration of each trial to a period of one hour. Of the 53 trials conducted, a very few were continued for a little less than one hour due to irregularities beyond control.

The engine-cylinder clearance was found by computing the volume of weight of water required to fill the clearance space. The engine was placed on dead-center by trammeling and the cylinder head and valves removed and heavy grease applied to the surfaces to prevent possible leakage. When all parts were properly assembled the clearance volume was filled with water through a hole in the top of the cylinder casting, its weight and temperature being taken. Both the head- and crank-end clearances

TABLE 2 AVERAGE OBSERVED AND COMPUTED RESULTS, SATURATED STEAM

	Exhaust condition: 21 in. vacuum						Atmospheric exhaust						
1 Trial No.	7	8	24	9	10	11	28	29	30	31	32	35	34
2 Net brake load, lb.	200	300	400	500	600	700	100	200	300	400	500	600	700
3 Duration of trial, min.	60	60	60	60	60	45	60	60	60	60	60	60	60
4 Steam-line pressure, lb. per sq. in. gage	133.62	131.9	134.15	130.8	133.3	133.4	133.46	133.77	136.69	137.1	133.31	136.46	140.0
5 Barometric pressure, in. hg	28.95	29.42	28.98	29.42	29.37	29.37	29.20	29.20	29.28	29.26	29.24	29.21	29.19
6 Revolutions per min.	219.1	218.4	214.3	213.5	211.6	209.1	219.0	214.85	214.7	211.6	208.9	207.6	205.7
7 Calorimeter pressure, lb. per sq. in. gage	4.98	5	5	4.98	5	5	5	5	5	5	5	5	5
8 Calorimeter temperature, deg. fahr.	274.8	270.3	272.5	269.5	275.5	276.4	277.8	279.3	278.8	279.5	279.2	279.4	278.5
9 Room temperature, deg. fahr.	90.5	83.8	88.5	83.8	85.0	89.1	85.7	85.8	79.2	81.9	83.8	90.3	92.3
10 Temp. of steam at throttle, deg. fahr.	357.5	357.7	357.8	356.7	357.7	358.1	355.8	355.9	357.2	356.9	354.8	356.5	357.4
11 Temp. of steam in jackets, deg. fahr.	354.0	353.4	354.9	353.4	355.1	355.2	349.6	349.6	350.1	350.0	350.9	351.9	351.8
12 Temp. of steam in exhaust, head end, deg. fahr.	152.7	155.5	152.5	156.7	156.1	158.6	212.8	212.2	214.5	215.0	215.0	215.0	215.0
13 Temp. of steam in exhaust, crank end, deg. fahr.	151.6	154.9	151.5	155.7	156.0	158.0	215.0	215.0	211.9	212.0	212.0	212.0	213.0
14 Pressure in exhaust, lb. per sq. in. gage							0	0	0	0	0	0	0
15 Vacuum in condenser, in. hg	21.15	21.1	21.01	21.06	21.03	20.91							
16 Steam used per hour from condenser, lb.	911.5	1267.5	1578.5	1966.0	2332.75	2718.0	820.0	1146.0	1550.5	1891.0	2282.0	2718.0	3190.5
17 Steam used per hour from jackets, lb.	93.0	109.0	116.5	125.0	124.5	133.3	95.0	105.5	126.0	124.0	128.0	137.5	143.0
18 Steam-line pressure, lb. per sq. in. abs.	147.86	146.35	148.38	145.25	147.73	147.83	147.80	148.11	151.07	151.47	147.67	150.81	154.34
19 Quality of steam at throttle, per cent	98.21	98.02	98.12	97.99	98.27	98.32	98.40	98.48	98.39	98.48	98.49	98.44	98.34
20 Total dry steam used per hour, lb.	986.52	1349.25	1663.13	2048.97	2414.74	2803.4	900.4	1232.48	1649.51	1984.37	2373.6	2810.95	3278.16
21 Mean effective pressure, head end, lb. per sq. in.	19.70	29.53	38.25	46.76	55.20	63.56	14.62	21.68	32.09	39.39	47.41	54.48	62.71
22 Mean effective pressure, crank end, lb. per sq. in.	17.13	23.86	30.56	37.04	42.68	49.34	5.76	13.5	20.86	27.72	34.13	41.69	48.45
23 Indicated horsepower, head end	30.39	45.42	57.72	70.29	82.25	93.59	22.55	32.80	48.52	58.69	69.75	79.65	90.84
24 Indicated horsepower, crank end	25.90	35.95	45.19	54.57	62.31	71.19	8.70	20.01	30.90	40.47	49.21	59.72	68.77
25 Total indicated horsepower	56.29	81.37	102.91	124.86	144.56	164.7	31.25	52.81	79.42	99.16	118.96	139.36	159.61
26 Dry steam used per i.hp.-hr.:													
By cylinder, lb.	15.91	15.27	15.05	15.43	15.85	16.22	25.82	21.37	19.21	18.78	18.89	19.20	19.66
By jackets, lb.	1.62	1.31	1.11	0.98	0.85	0.79	2.99	1.97	1.56	1.23	1.06	0.97	0.88
Total, lb.	17.53	16.58	16.16	16.41	16.70	17.01	28.81	23.34	20.77	20.01	19.95	20.17	20.54
27 Efficiency ratio, per cent	58.9	63.4	64.3	64.2	62.7	61.8	52.5	61.1	71.4	74.4	76.2	74.0	72.0

	Exhaust condition: 3 lb. back pressure							7 lb. back pressure					
1 Trial No.	15	17	16	18	19	20	22	14	33	13	12	36	21	23
2 Net brake load, lb.	100	200	300	400	500	600	700	100	200	300	400	500	600	700
3 Duration of trial, min.	60	60	60	60	60	60	60	60	60	60	60	60	60	60
4 Steam-line pressure, lb. per sq. in. gage	131.62	130.15	128.08	135.0	136.0	133.23	134.23	130.92	136.38	135.46	134.15	134.92	133.69	133.54
5 Barometric pressure, in. hg	29.27	29.14	29.13	26.6	29.6	29.31	29.24	29.27	29.22	29.23	29.25	29.21	29.23	29.23
6 Revolutions per min.	217.6	214.6	211.7	210.2	208.4	206.9	205.1	215.4	212.0	210.8	208.8	206.7	205.4	202.8
7 Calorimeter pressure, lb. per sq. in. gage	5	5	5	5	5	5	5	5	5	5	5	5	5	5
8 Calorimeter temperature, deg. fahr.	271.8	273.1	272.6	273.2	273.9	276.0	274.7	271.3	279.3	279.6	277.8	279.0	277.0	269.9
9 Room temperature, deg. fahr.	88.5	84.0	82.9	83.2	83.7	82.2	85.9	86.4	92.8	90.2	88.5	91.5	82.6	86.8
10 Temp. of steam at throttle, deg. fahr.	357.7	356.3	355.2	358.9	359.2	357.8	357.8	357.1	357.9	358.9	358.0	356.5	358.6	357.8
11 Temp. of steam in jackets, deg. fahr.	354.2	353.2	351.6	355.8	356.4	354.8	354.6	353.6	353.2	356.1	355.8	352.5	355.4	354.8
12 Temp. of steam in exhaust, head end, deg. fahr.	227.3	223.0	223.1	224.0	224.0	223.0	223.8	237.8	234.2	234.1	233.5	233.2	233.6	233.0
13 Temp. of steam in exhaust, crank end, deg. fahr.	224.5	220.0	219.8	222.0	222.0	220.5	221.0	234.2	230.8	229.9	227.9	230.2	230.2	230.1
14 Pressure in exhaust, lb. per sq. in. gage	3	3	3	3	3	3	3	7.05	7	7.08	7	7	7	7
15 Vacuum in condenser, in. hg														
16 Steam used per hour from condenser, lb.	872.5	1228.0	1603.5	1998.0	2404.0	2862.0	3359.0	999.0	1342.5	2002.25	2155.75	2536.0	3044.5	3580.0
17 Steam used per hour from jackets, lb.	94.5	117.0	120.0	131.0	137.5	145.5	163.0	98.5	129.0	135.0	133.75	140.0	153.0	167.0
18 Steam-line pressure, lb. per sq. in. abs.	146.0	144.46	142.39	149.54	150.54	147.63	148.59	145.30	150.73	149.82	148.52	149.26	148.05	147.90
19 Quality of steam at throttle, per cent	98.09	98.19	98.2	98.12	98.15	98.33	98.22	98.08	98.44	98.48	98.38	98.44	98.37	98.00
20 Total dry steam used per hour, lb.	948.53	1320.66	1692.48	2088.97	2494.48	2957.27	3459.31	1076.42	1448.44	2104.76	2252.41	2634.25	3145.38	3672.1
21 Mean effective pressure, head end, lb. per sq. in.	15.3	22.62	31.65	41.07	48.60	54.96	62.45	15.40	23.45	32.07	40.85	45.96	54.74	61.98
22 Mean effective pressure, crank end, lb. pr sq. in.	6.77	15.17	21.41	27.28	33.80	40.55	46.48	6.16	12.26	20.23	26.54	32.73	40.71	46.25
23 Indicated horsepower, head end	23.44	34.18	47.19	60.79	71.32	80.08	90.20	23.37	34.92	47.61	60.06	66.90	79.18	88.51
24 Indicated horsepower, crank end	10.17	22.46	31.27	39.57	48.81	57.89	65.78	9.16	17.93	29.42	38.24	46.68	57.70	64.72
25 Total indicated horsepower	33.61	56.64	78.46	100.36	120.13	137.97	155.98	32.52	52.85	77.03	98.30	113.58	136.88	153.23
26 Dry steam used for i.hp.-hr.:														
By cylinder, lb.	25.46	21.29	20.07	19.53	19.64	20.40	21.15	30.13	25.01	25.59	21.57	21.97	21.88	22.89
By jackets, lb.	2.76	2.03	1.50	1.28	1.12	1.03	1.03	2.97	2.40	1.73	1.34	1.21	1.10	1.07
Total, lb.	28.22	23.32	21.57	20.81	20.76	21.43	22.18	33.10	27.41	27.32	22.91	23.18	22.98	23.96
27 Efficiency ratio, per cent	58.0	70.4	76.9	77.9	77.6	76.3	73.3	54.1	64.2	64.8	77.7	75.9	77.4	74.3

were found in this manner. Four trials were made on each end, the results of which checked very very closely. The average results found are given in Table 1.

THE RESULTS

The average observed and computed results of the different trials as described above are presented in Tables 2 and 3. Calibrations of apparatus such as pressure gages, weighing scales, thermometers, indicator springs, etc., were made, and where corrections were found these have been included in the results presented.

The more important items of these tables are presented graphically in Figs. 3 to 6. In plotting these curves the indicated horsepower was used as a common abscissa. The effect upon the steam consumption and efficiency ratio is clearly shown.

As the trials progressed it was found that the condenser circulating-water supply was insufficient to maintain a vacuum of 26 in. of mercury under all load conditions. For this reason it became necessary to maintain as nearly as possible a vacuum of 21 in. for all load conditions to facilitate comparisons. Time did not permit of other trials at a vacuum less than 21 in., from which it might be possible to estimate the steam consumption at a 26-in. vacuum. Since it seemed desirable to estimate the dry-steam consumption at a vacuum of 26 in. the following method was used:

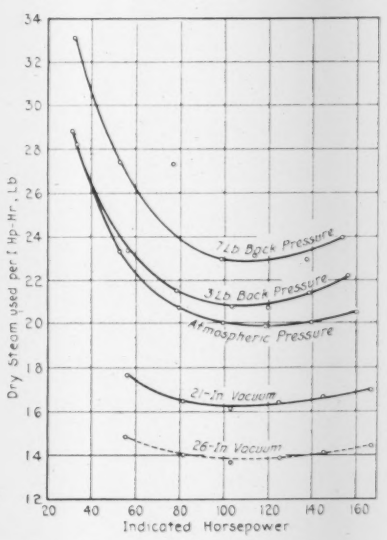

FIG. 3 WATER-RATE CURVES, SATURATED STEAM

TABLE 4 AVERAGE EFFICIENCY RATIOS FOR DIFFERENT EXHAUST PRESSURES

Item	Exhaust pressure	Efficiency ratio: Saturated steam	Superheated steam
1	21 in. vacuum = 4.00 lb. per sq. in. abs.	0.626	0.658
2	Atmosphere = 14.38 lb. per sq. in. abs.	0.62	0.774
3	3 lb. gage = 17.38 lb. per sq. in. abs	0.726	0.767
4	7 lb. gage = 21.38 lb. per sq. in. abs.	0.693	0.761
5	26 in. vacuum (estimated) = 1.62 lb. per sq. in. abs.	0.613	0.638

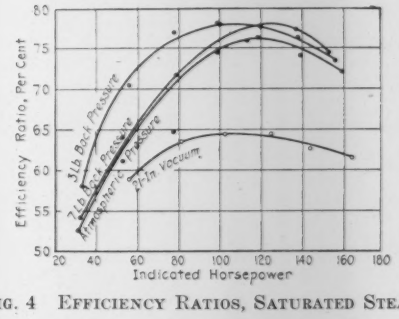

FIG. 4 EFFICIENCY RATIOS, SATURATED STEAM

The average efficiency ratio based upon the indicated horsepower was computed for each exhaust condition, the results secured being presented in Table 4. In this table the first four items are those found from test conditions. These four items are plotted in Fig. 7 in order to determine the relation between the efficiency ratio and exhaust pressure. After finding the direction of the line it was extended to include the 26-in. vacuum, the results

TABLE 3 AVERAGE OBSERVED AND COMPUTED RESULTS, SUPERHEATED STEAM

	Exhaust condition											
	21 in. vacuum						Atmospheric exhaust					
1 Trial No.	37	26	25	27	38	39	40	41	42	43	44	45
2 Net brake load, lb	250	300	400	500	600	700	200	300	400	500	600	700
3 Duration of trial, min	60	60	60	60	60	60	60	60	60	60	55	68
4 Steam-line pressure, lb. per sq. in. gage	132.15	133.92	133.62	134.15	126.92	133.23	135.38	130.62	127.46	124.15	131.77	136.69
5 Barometric pressure, in. hg	29.38	29.14	28.99	29.14	29.38	29.37	29.35	29.46	29.44	29.45	29.46	29.28
6 Revolutions per min	217.95	217.5	214.5	213.0	210.2	208.5	215.5	214.2	210.2	208.5	208.5	207.4
7 Room temperature, deg. fahr	87.9	86.8	89.2	87.8	89.5	92.5	94.2	91.8	92.4	91.9	92.0	84.7
8 Temp. of steam at throttle, deg. fahr	471.2	466.8	463.0	454.7	457.2	457.2	464.3	464.1	467.2	464.0	471.5	461.5
9 Temp. of steam in jackets, deg. fahr	351.0	350.0	349.2	350.8	351.2	350.2	350.4	351.8	349.5	348.9	353.8	356.5
10 Temp. of steam in exhaust, head end, deg. fahr	153.8	153.4	152.9	160.0	156.7	157.7	215.1	214.9	215.0	214.7	214.9	215.0
11 Temp. of steam in exhaust, crank end, deg. fahr	153.0	152.4	152.2	158.2	155.9	156.7	213.2	213.3	212.9	212.9	213.3	212.8
12 Pressure in exhaust, lb. per sq. in. gage							0	0	0	0	0	0
13 Vacuum in condenser, in. hg	20.98	21.12	21.08	20.91	20.92	21.07						
14 Steam used per hour, from condenser, lb	962.0	1133.0	1437.5	1746.0	2087.0	2439.5	1026.5	1392.5	1711.5	2085.0	2395.6	2860.5
15 Steam used per hour, from jackets, lb	54.5	61.5	40.0	64.0	58.5	65.0	56.5	57.0	60.5	60.5	62.2	71.5
16 Steam-line pressure, lb. per sq. in. abs	146.58	148.23	147.86	148.46	141.35	147.66	149.81	145.09	141.92	138.62	146.24	151.07
17 Degrees superheat at throttle, deg. fahr	114.5	109.2	105.63	97.02	103.4	99.9	105.9	108.2	113.0	111.7	115.0	102.4
18 Total dry steam used per hour, lb	1016.5	1194.5	1477.5	1810.0	2145.5	2504.5	1083.0	1449.5	1772.0	2145.5	2457.8	2932.0
19 Mean effective pressure, head end, lb. per sq. in.	24.84	30.31	39.92	45.77	55.85	62.68	22.51	32.88	40.08	48.01	55.23	65.37
20 Mean effective pressure, crank end, lb. per sq. in.	20.28	21.77	29.37	34.93	42.64	48.71	14.40	20.87	27.36	34.23	39.97	47.41
21 Indicated horsepower, head end	38.12	46.42	60.30	68.65	82.67	92.03	34.16	49.60	59.33	70.49	81.09	95.47
22 Indicated horsepower, crank end	30.49	32.67	43.47	51.34	61.84	70.08	21.41	30.84	39.68	49.24	57.49	67.86
23 Total indicated horsepower	68.61	79.09	103.77	119.99	144.51	162.11	55.57	80.44	99.01	119.73	138.58	163.32
24 Dry steam used per i.hp.-hr.:												
By cylinder, lb	14.02	14.32	13.85	14.55	14.45	15.05	18.47	17.31	17.29	17.42	17.29	17.51
By jackets, lb	0.80	0.78	0.39	0.53	0.40	0.40	1.02	0.71	0.61	0.50	0.45	0.44
Total, lb	14.82	15.10	14.24	15.08	14.85	15.45	19.49	18.02	17.90	17.92	17.74	17.95
25 Efficiency ratio, per cent	64.8	64.1	68.0	65.2	66.9	63.7	71.2	78.0	78.3	80.0	78.4	76.6

	Exhaust condition													
	3 lb. back pressure							7 lb. back pressure						
1 Trial No.	46	47	48	49	50	59	58	51	52	53	54	55	56	57
2 Net brake load, lb	100	200	300	400	500	600	700	100	200	300	400	500	600	700
3 Duration of trial, min	60	60	60	60	60	60	60	60	60	60	60	60	60	60
4 Steam-line pressure, lb. per sq. in. gage	138.54	138.15	135.38	136.69	136.38	133.92	134.0	132.62	133.69	133.92	136.00	134.38	134.00	133.46
5 Barometric pressure, in. hg	29.31	29.37	29.38	29.39	29.4	29.15	29.13	29.64	29.62	28.95	28.99	29.00	29.17	29.14
6 Revolutions per min	217.4	214.9	211.6	210.25	208.3	206.5	204.9	214.9	212.4	210.6	209.3	207.1	205.9	203.25
7 Room temperature, deg. fahr	87.0	89.8	90.3	92.9	93.8	99.4	97.8	79.8	83.7	95.7	97.4	95.5	94.5	97.4
8 Temp. of steam at throttle, deg. fahr	469.5	460.4	460.5	464.0	454.5	469.0	463.8	469.7	466.3	459.2	468.2	454.7	456.8	455.4
9 Temp. of steam in jackets, deg. fahr	357.6	357.6	356.4	356.8	356.5	355.5	355.6	355.0	355.2	356.2	357.0	355.8	355.8	355.5
10 Temp. of steam in exhaust, head end, deg. fahr.	248.5	225.4	224.5	224.7	224.0	224.1	224.0	260.1	237.7	234.8	234.2	234.1	234.0	234.1
11 Temp. of steam in exhaust, crank end, deg. fahr.	248.4	225.0	221.0	221.0	220.2	221.2	221.2	264.4	240.2	231.8	231.2	231.1	231.0	231.0
12 Pressure in exhaust, lb. per sq. in. gage	3	3	3	3	3	3	3	7	7	7	7	7	7	7
13 Vacuum in condenser, in. hg														
14 Steam used per hour, from condenser, lb	792.5	1100.0	1408.0	1763.5	2163.0	2553.5	2910.5	908.75	1208.5	1586.0	1910.0	2285.5	2717.5	3107.5
15 Steam used per hour, from jackets, lb	46.0	58.0	61.5	63.0	62.5	59.5	65.0	43.5	51.0	65.0	58.5	60.5	72.5	69.0
16 Steam-line pressure, lb. per sq. in. abs	152.94	152.58	149.81	151.13	150.82	148.25	148.30	147.18	148.24	148.14	150.24	148.62	148.33	147.77
17 Degrees superheat at throttle, deg. fahr	109.5	100.6	102.1	104.9	95.6	111.4	106.2	112.7	108.7	101.7	109.6	96.9	99.2	98.1
18 Total dry steam used per hour, lb	838.5	1158.0	1469.5	1826.5	2225.5	2613.0	2975.5	952.25	1259.5	1651.0	1968.5	2346.0	2790.0	3176.5
19 Mean effective pressure, head end, lb. per sq. in.	15.6	24.04	31.70	40.07	47.88	56.55	63.58	16.59	24.93	33.77	39.96	47.24	56.26	62.72
20 Mean effective pressure, crank end, lb. per sq. in.	6.96	14.77	20.95	27.35	34.67	41.90	48.09	6.90	12.83	20.59	25.74	33.81	39.86	47.36
21 Indicated horsepower, head end	23.88	36.38	47.24	59.33	69.82	82.23	91.74	25.11	37.29	50.02	58.90	68.89	81.57	89.76
22 Indicated horsepower, crank end	10.44	21.90	30.59	39.68	49.83	59.70	67.99	10.23	18.80	29.92	37.17	48.31	56.63	66.42
23 Total indicated horsepower	34.32	58.28	77.83	99.01	119.65	141.93	159.73	35.34	56.09	79.94	96.07	117.20	138.20	156.18
24 Dry steam used per i.hp-hr.:														
By cylinder, lb	23.09	18.87	18.09	17.81	18.11	17.99	18.22	25.71	21.54	19.84	19.88	19.50	19.66	19.90
By jackets, lb	1.34	1.00	0.79	0.64	0.52	0.42	0.41	1.23	0.91	0.81	0.61	0.52	0.53	0.44
Total, lb	24.43	19.87	18.88	18.45	18.63	18.41	18.63	26.94	22.45	20.65	20.49	20.02	20.19	20.34
25 Efficiency ratio, per cent	60.4	75.3	80.2	80.7	81.1	81.3	80.8	60.6	72.7	78.3	78.6	82.3	81.3	81.1

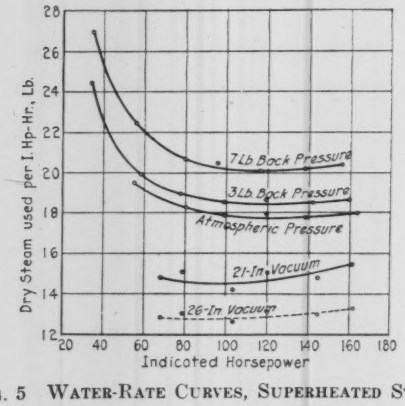

FIG. 5 WATER-RATE CURVES, SUPERHEATED STEAM

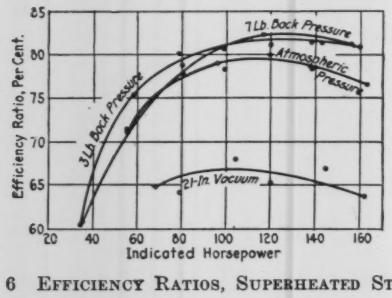

FIG. 6 EFFICIENCY RATIOS, SUPERHEATED STEAM

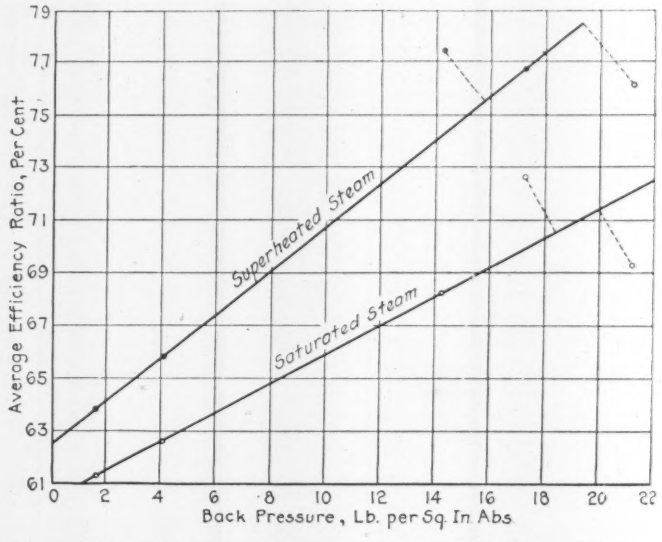

FIG. 7 RELATION BETWEEN EFFICIENCY RATIO AND BACK PRESSURE

secured being given in Table 4 as item No. 5. Having found the estimated average efficiency ratio for a vacuum of 26 in., the following equation was used in correcting the steam consumption from a vacuum of 21 in. to a vacuum of 26 in.:

Pounds dry steam consumed per i.hp-hr. at 26 in. vacuum

$$= \left(\frac{(H_1 - H_2)\,(\text{at 21 in. vac.})}{(H_1 - H_2)\,(\text{at 26 in. vac.})} \right) \times \left(\frac{\text{avg. eff. ratio for 21 in. vac.}}{\text{avg. eff. ratio for 26 in. vac.}} \right)$$

$$\times \text{(Lb. dry steam consumed i.hp-hr. at 21 in. vac.)}$$

where H_1 = total heat contents per pound of steam at the initial condition

H_2 = total heat contents per pound of steam at the final condition, assuming adiabatic expansion.

The application of the above equation gave the results presented in Table 5, which are shown as a dotted line on Figs. 3 and 5.

Sample indicator cards are presented as Figs. 8 and 9.

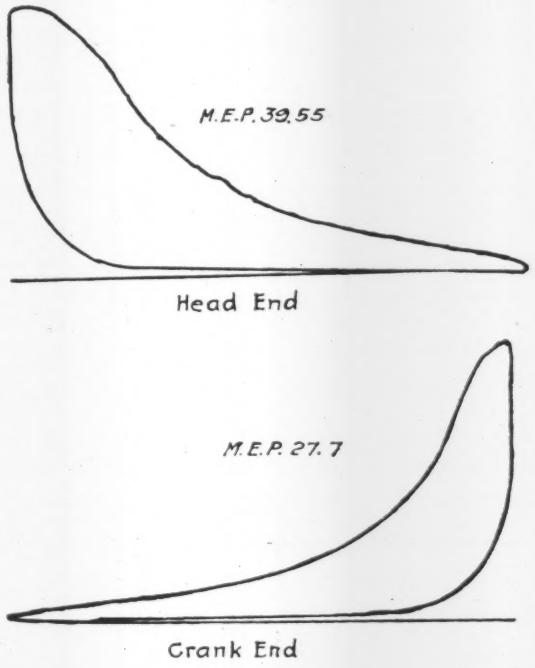

FIG. 8 TYPICAL INDICATOR CARDS, SATURATED STEAM
(400 lb. net load; all jackets)

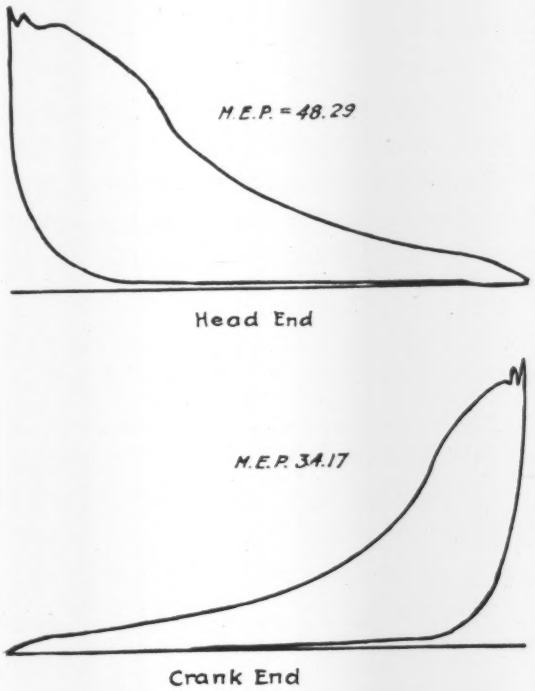

FIG. 9 TYPICAL INDICATOR CARDS, SUPERHEATED STEAM
(500 lb. net load; all jackets)

TABLE 5 ESTIMATED DRY STEAM USED PER INDICATED HORSEPOWER AT 26 IN. VACUUM—POUNDS

Saturated steam			Superheated steam		
Net brake load, lb.	Dry steam per i.hp-hr. 21-in. vac.	Dry steam per i.hp-hr. 26-in. vac.	Net brake load, lb.	Dry steam per i.hp-hr. 21-in.vac.	Dry steam per i.hp-hr. 26-in. vac.
200	17.53	14.87	250	14.82	12.77
300	16.58	14.03	300	15.10	13.01
400	16.16	13.68	400	14.24	12.61
500	16.41	13.90	500	14.85	13.10
600	16.70	14.12	600	15.08	12.97
700	17.01	14.40	700	15.45	13.28

Discussion

J. G. PROSSER[1] stated that the engine under consideration operated on the dry-steam cycle, which under average conditions of operation had an efficiency of 95 to 97 per cent of the Rankine wet-steam, or adiabatic, cycle. Further, while it was impossible to secure the true adiabatic in any cylinder, due to the heat interchange between the walls and the working vapor, it had been possible to attain from 82 to 92 per cent of the wet-steam-curve efficiency by working on the dry-steam principle. These economies were obtained after considerable development. Speed limitations had been encountered owing to the difficulty of seating poppet valves without undesirable hammering on the seat, unless a cam of very easy angle was used, which gave a wire-drawn admission or else a wire-drawn cut-off. Piston valves were employed to avoid this difficulty, and speed was not now limited by mechanical conditions. The engine showed up particularly well with high back pressures, as where the exhaust was used for process work; the exhaust lap on the main valve in such cases could be so fixed as to prevent looping at the top, while the compression remained unchanged for all loads and at any selected point, hence giving a very flat and economical steam-consumption curve.

E. H. Lockwood[2] thought it would be interesting to compare the steam consumption of this completely jacketed engine with that of other engines of the same speed and size, but unjacketed, of the uniflow type. He criticized omission of mention of steam used to drive the auxiliaries. Such steam was properly chargeable to the engine and might easily add 10 per cent to the total steam consumption for the condensing runs. Professor Lockwood was also interested in another unreported item, the brake horsepower developed, which by comparison with the indicated horsepower would give the friction losses. In the Prosser engine the use of two additional stuffing boxes made friction loss of particular interest. He also asked whether the tests with superheated steam included any runs with no steam in the jacket, so as to deduce the effect of jackets when used with superheated steam.

W. Turnwald[3] was interested in knowing why the curves of steam consumption against indicated horsepower showed such an increase in lb. per i.hp-hr. with atmospheric and back pressures for light loading (as in Figs. 3 and 5), while the runs with partial vacuum showed no such increase.

J. T. Wilkin[4] asked for a restatement of the percentage of jacket steam used.

F. W. Dean[5] pointed out that since the author had had difficulty in keeping the power constant with the brake, and admitted his brake horsepower to be somewhat uncertain, his indicated horsepower might be uncertain for the same reason.

Jos. J. Nelis[6] asked what economies might be effected with the Prosser-type engine for marine work, with a standard horsepower and constant load of between 2500 and 3000. Mr. Prosser answered this question by saying that his company had never built engines for marine work and that he could only express an opinion. In the marine engine when the steam passed through several cylinders, there were decided advantages over a single cylinder from a thermal standpoint. For this reason his company would expect at least as high a thermal efficiency with a marine engine using a single piston valve of the type now in common use.

H. G. Mueller[7] submitted an interesting written discussion in which he said that innumerable tests on both counterflow and uniflow-type steam engines had shown that the Rankine-cycle ratio dropped off rapidly as the vacuum was increased, and also that the curve reached its highest value with moderate back pressures, such as were commonly used for process steam in industrial plants, varying from atmospheric to 10 lb. gage, and sometimes 15 to 20 lb. gage.

Fig. 7 of the paper was therefore incorrectly drawn. The curves

[1] Prosser Co., Chicago, Ill. Mem. A.S.M.E.
[2] Professor of Mechanical Engineering, Sheffield Scientific School, Yale University. Mem. A.S.M.E.
[3] Consulting Engineer, Syracuse, N. Y. Mem. A.S.M.E.
[4] Pres., Connersville Blower Company, Connersville, Ind. Mem. A.S.M.E.
[5] Wheelock, Dean & Bogue, Inc., Boston, Mass. Mem. A.S.M.E.
[6] Power Specialty Co., New York, N. Y. Mem. A.S.M.E.
[7] Chief Engineer, Skinner Engine Co., Erie, Pa.

should not be straight lines with the efficiency increasing in direct proportion to the back pressure, but should be flattened out as the back pressure increased and dropped very materially for the high vacua. The points obtained from averaging the results would indicate curves of this nature rather than the straight lines shown by the author.

The principal reason for believing that this should be the character of the curve was the rapid increase of the B.t.u. available in the Rankine cycle with increased vacua. It seemed logical to think that the engine would extract a lesser portion of the B.t.u. available as the available heat increased. To illustrate this Mr. Mueller had plotted Fig. 10, in which was indicated, for both saturated- and superheated-steam tests, the average steam consumption

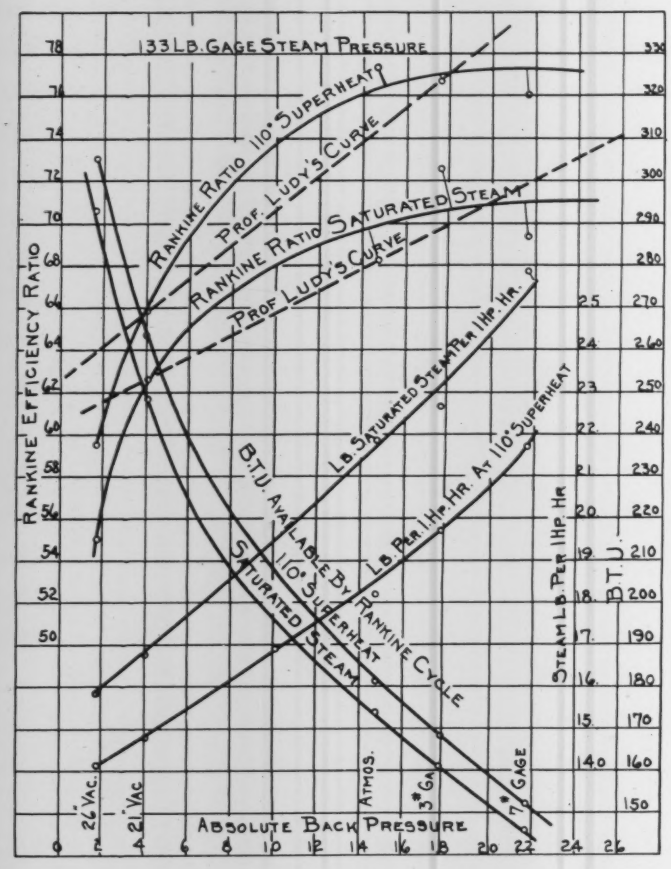

FIG. 10

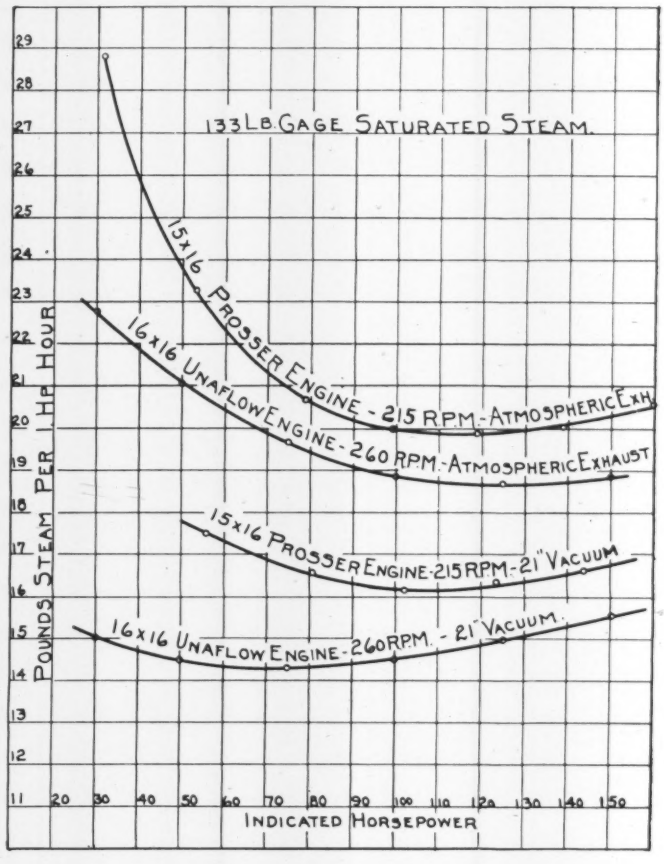
133 LB. GAGE SATURATED STEAM
15x16 PROSSER ENGINE - 215 RPM - ATMOSPHERIC EXH.
16x16 UNAFLOW ENGINE - 260 RPM - ATMOSPHERIC EXHAUST
15x16 PROSSER ENGINE - 215 RPM - 21" VACUUM
16x16 UNAFLOW ENGINE - 260 RPM. - 21" VACUUM.
POUNDS STEAM PER 1 HP HOUR
INDICATED HORSEPOWER

FIG. 11

in pounds per i.hp-hr. at various back pressures from 21 in. vacuum to 7 lb. gage, and with projections to 26 in. vacuum; also the B.t.u. available by the Rankine cycle at these various back pressures, and the Rankine-cycle ratios as averaged from the tests and with curves projected to 26 in. vacuum. In comparison with these he had drawn in Professor Ludy's straight lines.

From this comparison it was interesting to note that the correction for 26 in. vacuum over 21 in. vacuum averaged about 0.8 lb. of steam per i.hp-hr. for the saturated-steam tests and about 0.9 lb. for the superheated-steam tests. This was in contrast to the correction used by the author which averaged about 2.0 lb. per i.hp-hr. It was therefore probable that this engine when running with 26 in. vacuum would show a steam consumption averaging around 15.9 lb. with saturated steam and 13.9 lb. with superheated steam.

With 4.6 per cent clearance and the short compression stroke shown by the non-condensing cards in Figs. 8 and 9, it was quite possible that the engine would show even less improvement in steam consumption with 26 in. vacuum over 21 in. vacuum than estimated above, due to the fact that the compression would be exceedingly low and require excessive lead of the steam valve to produce an indicator card in which the initial pressure would be a reasonable proportion of the throttle pressure.

No condensing cards were shown in the paper, and no tests were run at 26 in. vacuum. It might be assumed that the high vacuum cards were not well proportioned and this might have been one of the reasons for running with a lower vacuum. This condition, no doubt, had been considerably aggravated by the type of valve gear and governor used, which did not give a good distribution of the load on either end of the cylinder as evidenced by the very large difference in the mean effective pressures between the two ends of the cylinder, this difference changing as the load was varied, and always carrying the heavier load on the head end. This was a characteristic defect of a slide steam valve on an engine with a short connecting rod.

The condensing curves plotted in Figs. 3 and 5 for a constant steam condition against variable loads were no doubt typical for this type of engine, but were not typical of the more efficient uniflow-type engines commonly built today, in which the steam consumption improved as the load dropped off, down to a very low mean effective pressure, which approached the mean effective pressure obtained on the Rankine cycle for the same condensing operating condition. In other words, the more efficient uniflow engine operated closer to the ideal Rankine cycle than did the Prosser-type engine.

Professor Ludy in his paper had listed methods commonly employed to obtain ideal expansion in which no heat was lost as heat, but had not mentioned the uniflow principle. This, combined with steam-tight poppet valves, small clearances, and short compression strokes obtained when operating non-condensing by auxiliary exhaust valves located so as to prevent the cooler exhaust from washing the hot steam jackets, was commonly used today. The application of these principles, together with a cylinder properly proportioned, made possible, without the use of steam-heated pistons and their complications, and without compounding cylinders, economies considerably better than those obtained by the Prosser principle. Furthermore the application of these principles not only improved the economies at the heavy loads but even more so at the lighter loads, and these economies were maintained throughout the life of the engine. The uniflow engine on account of more nearly approaching the Rankine cycle than a counterflow engine, required a cylinder of greater piston displacement for

producing the same horsepower with best economy. In Fig. 11 such a comparison was made between a 16- by 16-in. well-known uniflow engine running at 260 r.p.m. and the 15- by 16-in. Prosser engine at 215 r.p.m.

It was a well-known fact that piston valves had been discarded by the builders of high-class engines, not only in Europe but in this country, because they suffered heavy steam leakage as wear took place, and required frequent renewals of their component parts for maintained steam-tightness: whereas a properly designed poppet valve when new would show steam escapement under full pressure difference of less than $^1/_2$ of 1 per cent of the steam used at full-load rating, and this escapement diminished with continued use of the engine. The Prosser engine under discussion was undoubtedly a new unit, carefully fitted in all details, and after one or two years' use the steam consumption might be materially increased, especially at the lighter loads where the valve leakage was greater in proportion to the total steam used.

The steam-heated piston had also been discarded by a uniflow-engine builder, for the reason that comparative tests had shown the steam consumption with and without the live steam inside of the piston, to be so small an improvement that it did not warrant the additional complications in the tail-rod construction and tail-rod packing, which were necessary to insure proper circulation and drainage of condensation. The construction used on the Prosser engine under discussion not only produced a considerable bending moment on the piston rod due to the fact that the tail rod and piston rod were not concentric, but also the check-valve construction and method of supplying steam to the piston head required that the hollow tail rod be about half-full of water before any of the condensation was removed.

The above-mentioned uniflow-engine builder had also found in his experiments with steam-heated pistons that check valves were necessary both in the tail rod and its housing, using the tail rod as a plunger to obtain proper drainage and circulation. Even then the improvement in economy was very small. The tail rod was made concentric with the piston rod to prevent a bending moment on either, which might cause serious packing troubles with usage and to eliminate large running clearances which would be necessary on an eccentric construction such as used by the Prosser engine, due to the possibility of the piston not riding parallel with the bore of the cylinder.

Attention was called to the Armstrong governor employed to operate the piston valve, which required a large difference in travel from no load to full load, and a correspondingly large difference in eccentricity of the steam eccentric, resulting in poor speed regulation. The tests showed a drop of 9 c. 10 revolutions between the loads used, or approximately 5 per cent variation, which would not be acceptable for installations where heavy surges in the load existed, due to elevators, cranes, or other large motors on the line with heavy starting torques, or where a voltage drop was objectionable with load change.

Professor Ludy closed the discussion. In dealing with Mr. Turnwald's inquiry regarding steam consumption at light loading, he pointed out that owing to a constant valve setting it was rather difficult to obtain definite values in this region. Answering Mr. Wilkin, he gave a restatement of the saving effected by the use of jacketing as follows: Special tests to determine these values were made at a brake load of 400 lb. when using saturated steam and 500 lb. when using superheated steam. The saving was given in per cent of the steam consumed per indicated horsepower per hour when no jackets were used. For saturated steam, the saving when using all jackets was 11.94 per cent and when using the piston and rod jacket only, was 3.48 per cent. For superheated steam the saving when using all jackets was 7.33 per cent, and when using the piston and rod jacket only, was 2.23 per cent.

With regard to brake horsepower and friction loss, the results obtained from the Prony brake, operated by hand, were at times irregular. An average of about 8 per cent represented the friction loss of the engine. With regard to the frequency with which observations had been taken, Professor Ludy said that they were made every five minutes and that most tests were one hour in duration.

Industrial Use of the Oil Engine

How One of the Factors in the Decentralization of Industry Is Affected by an Economical Source of Power in Small Units

BY L. H. MORRISON,[1] NEW YORK, N. Y.

A CENTURY ago industry was of the most primitive form. Each tiny hamlet had its group of artisans in metals, textiles, and foodstuffs, the work for the most part being carried on in the home. The perfection of the steam engine and the introduction of power-driven machinery changed the entire form of industrial life. In place of the fireside factory there came into existence the mill and shop system of today where in a single building thousands of hands attend machines, the output of one of which exceeds that of hundreds of the old-time artisans.

As industry expanded to meet the increased demands made possible by the machine age, the hamlet and village could not supply the necessary labor, and the factory was forced to shift the scene of operations to the more densely populated centers.

Still another incentive to the crowding of factories into the cities was the necessity of cheap power.

In recent years improvements in manufacturing methods have been such that production has exceeded consumption, and few shops are operating at 100 per cent capacity. At the same time, half of the population are going without many of the necessities of life simply because the consumer's cost is too high.

The present-day problem that must be solved before we can achieve real prosperity is that of a reduction of the cost to the ultimate consumer. Examination will reveal that the greatest obstacle to the attainment of this solution is the high cost of distribution. As long as freight rates, wages, and raw materials were low in price it little mattered where an object was manufactured. The larger centralized plant by the exercise of certain inherent advantages could manufacture far cheaper than could the small shop, and the retail price was little affected by a transportation movement of a thousand or so miles.

The concentration of factories in large industrial centers has served to increase the cost of living in such centers, which in turn has been reflected in increased manufacturing costs. Labor disturbances are likewise more prevalent in the densely populated districts.

If consumption is to be increased not only must manufacturing cost be reduced, but—of far greater importance—distribution must be accomplished more cheaply than at present.

Many industrial leaders agree that the decentralization of the factory system must take place.

By proper location of small factories in strategic points the entire country may be within a few hundred miles of the shop. Freight costs will be cut, and motor-truck transportation become a fact. Due to cheap living costs in these small towns, wages will cease to rise each year and labor disputes will largely disappear. In fact, several industries carried this plan into effect years ago, notably that of shoemaking, where a single manufacturer may have over a dozen branch factories. There is, unfortunately, a serious obstacle to the general adoption of this system.

Investigation reveals that, where decentralization has become a

[1] Assoc. Editor, *Power*, McGraw-Hill Co., Inc. Assoc-Mem. A.S.M.E.

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fact, the cost of power in the particular field represents but a minute percentage of the manufacturing expense. In truth, an industry requiring a large amount of power per unit of product, under existing circumstances cannot afford to adopt the small-factory plan. As long as steam power is necessary the factory must be large, for the efficiency of the steam power plant decreases more or less geometrically with a decrease in capacity.

It is the desirability of supplying cheap electric energy to the small factory as well as to the farm that has prompted the creation of the Giant Power Plan advocated by Governor Pinchot of Pennsylvania. The same desire for cheap energy was the basic motive for the Superpower Plan with its network of interconnecting lines. Just how much these two plans will assist decentralization is difficult of gaging. There is logic in the argument that the cost of energy should be more or less based on the distance this energy must be carried, and if this be true then industry will congregate in the vicinity of the power plant where the energy will be cheapest, at a considerable distance from the center of population.

The present system of rate making likewise favors the large factory, since it is only in the high blocks that the rate per kilowatt-hour becomes very low. Ultimately this may be altered to give the user, regardless of the installed capacity, the same proportional rates for the same load conditions as those obtainable by the large power consumer. When this is done, the building of small factories in thousands of villages will become an actuality.

Adaptability of the Oil Engine

In studying the power problem of the small factory the oil engine has been largely ignored. Due to its particular adaptability to oil-pipe-line and small-central-station use, its possibilities as a prime mover for industrial plants have not been appreciated, even by its most enthusiastic supporters.

In so far as cheap power is concerned, the Diesel engine is well worth the closest scrutiny. While the operating cost per unit generated in a steam plant increases rapidly with a decrease in capacity, the cost with a Diesel is practically constant regardless of capacity. The fuel consumption per horsepower-hour is the same in a 100-hp. as in a 3000-hp. unit; the same holds true in respect to the cost of lubrication and repairs. Whatever variation exists in the repair item is due to a difference in the character of the attendants, for it is too customary to employ the most unskilled labor in small plants from a mistaken sense of economy.

As compared to a steam plant of equal capacity, one is quite safe in stating that the oil engine will always prove the more efficient, including all charges, save where much hot water or process or heating steam is needed. In these exceptions the steam plant will usually but not always prove the more economical. In many instances the installation of waste-heat boilers and the recovery of the jacket water will provide ample hot water for the factory purposes. In fact, even in plants where steam-heating requirements are high during the winter, it is not only possible but probable that the installation of an oil engine for power, with the use of steam boilers for heating, will be the most economical solution. Operated in conjunction with an exhaust-heat boiler and using the jacket water for boiler feed, the overall efficiency of the Diesel portion of such a plant will be over 70 per cent. This is greater than the net boiler efficiency usually obtained in small plants. Without going into a mathematical proof, it may be stated that whenever the heating and power become unbalanced and the steam engine discharges its exhaust to the atmosphere, the oil-engine plan becomes the economical one. In other words, the oil engine would be as economical as the steam engine during the winter months, even though most of the steam heating be supplied by coal-fired boilers, and during the non-heating season its efficiency in terms of net cost per unit delivered at the switchboard would probably be twice that of the steam unit. The accuracy of these figures depends, of course, on the relative cost of coal and oil.

In the last two or three years unit capacities have increased to the point that units of 2000 to 3000 hp. are in successful operation. Several builders are prepared to construct even larger units, as the recent Shipping Board purchases prove. At least two builders are ready to supply 6000-hp. units, while in Europe engines of 6500 hp. capacity are in service and a 16,000-hp. 9-cylinder Diesel is in the course of construction. Since units of even 3000 hp. fulfil the needs of 99 per cent of industrial plants, for installations of two to five engines are not impractical, lack of capacity is not an obstacle to a more extended sphere of application.

In those sizes falling within what might be termed the industrial class, that is, engines of over 200 hp., the Diesel using air for injection purposes predominates; and as long as the units were held below 700 to 1000 hp., the four-stroke cycle held supremacy. This was not due to any inherent advantage over the two-stroke cycle, but was the result of unfortunate experiences with the early designs of the latter engine. Of American engines of over 1000 hp. the two-stroke cycle is in the majority.

While four-stroke-cycle single-acting engines of 2000 hp. are in operation, units of larger outputs must be either of the double-acting type, if operating on the four-stroke cycle, or the two-stroke cycle must be adapted.

The solid-injection engine, in which a mechanical oil pump is the sole injection agency, is the most popular unit in capacities under 300 to 400 hp. The development of this type holds peculiar interest, since designers in proceeding along three distinct lines have arrived at almost identical solutions. Some of these cold-starting engines are the outgrowths of improvements in the semi-Diesel or hot-bulb engines. By gradually increasing compression pressures and overcoming difficulties accompanying each improvement in compression, the semi-Diesel has emerged as a true heavy-oil engine of high economy. Most of these solid-injection engines are of the two-stroke cycle, and without exception these have incorporated in their design a precombustion chamber. In fact, one is quite safe in stating that some such limitation on the rate of combustion is necessary for all two-stroke-cycle engines where the scavenging and exhaust ports are in the same end of the cylinder.

Another school of designers have taken the four-stroke-cycle semi-Diesel or hot-surface engine as a foundation. Through gradual development a cold-starting engine has been evolved by a number of these designers. In all of these the combustion chamber of the semi-Diesel has been retained and a thorough mixture of the air and fuel brought about either through an elongated piston or by opposing oil sprays.

The third design can be traced to the air-injection Diesel. No combustion chamber other than the engine cylinder is used. The oil spray is introduced along the cylinder axis, and mixture is brought about by first vaporizing the oil by direct impingement on the hot piston crown, followed by a diffusion throughout the air mass. These engines are all four-stroke-cycle and may or may not be provided with mechanically actuated oil-spray needles.

The economy of the solid-injection engine is equal to that of the true Diesel. By reason of its cheaper construction, brought about through the elimination of the costly air compressor, it will ultimately force the Diesel out of the size range between 100 and 500 hp.

The semi-Diesel, in point of number and horsepower, has been and still is the favorite. Being in the main of the two-stroke-cycle type, its first cost is far below that of a more economical engine. The rugged construction and simplicity of design appeal to a vast number, and as long as oil is cheap, its rather high fuel consumption is offset by its advantages for installations of 150 hp. and under. There are over 1,000,000 hp. of these engines in service.

The factor that has heretofore prevented consideration of the Diesel, that is, the lack of large units, no longer exists. Since units of 6000 to 10,000 hp. are available, almost any plant capacity requirement can be met.

It is seldom that the oil engine is the subject of any discussion without the question of fuel supply cropping up. Governmental agencies are prone to issue oil-shortage warnings, or data which consist in the main of guesswork. No geologists of standing will venture to claim ability to define the limit of the new oil fields, or the amount of oil still underground. True, oil-production methods are wasteful and should be improved, but as far as is known, there will be a fuel supply for oil engines for years to come.

Even if the supply were to disappear in ten years, there are thousands of square miles of shale, representing billions of barrels of shale oil. Low-temperature distillation of coal is coming and will make available a large supply of tar oil, a most suitable Diesel fuel.

In conclusion, the author would repeat that cheap power is an industrial necessity, and engineers should by no means fail to include the oil engine in their studies of power sources.

Discussion

IN REPLY to a question asking how to start a Diesel engine when the air had been lost out of the bottles, the author said that there were several ways of starting. There should be a small stand-by compressor and gasoline engine to take care of losses of air from the air bottles. However, it was always possible to get a couple of bottles of carbon dioxide at a soda fountain, which could be used in an emergency.

In nearly every plant there was some method of getting air at 100 lb. pressure. If such air was available, the bottles should be charged with it and then heated to a not too high temperature, which would increase the pressure. This would provide air for starting. It was not necessary to have high pressure for starting; 200 lb. was sufficient if the valve gear was relieved on all cylinders except the one with the air-starting rig. The machine could be turned over idle, and after the flywheel had acquired a little momentum, one of the valve gears could be engaged, and as soon as that cylinder was firing, the same thing could be done with the others.

It was good practice to start with kerosene to clean out the spray valves, and to run for a few moments on kerosene when stopping. The cost was insignificant and the fuel valves were kept free of tar. It was sometimes possible to use the alternator as a synchronous motor, provided power was available, and start the engine by means of it.

E. D. Dreyfus[1] said that the factor of greatest importance in the use of the oil engine was the ultimate cost over a period of time. He recalled an experience in an electric station which contained eight or ten 1250-kw. natural-gas units. The station averaged for a short time about two mills per kilowatt-hour, all told, fuel, labor and repairs. However, as the supply of natural gas began to diminish and the prices went up, and as the engines were seeing harder and harder service, this very low cost advanced to about two cents per kilowatt-hour, an increase of about ten times, and the station was abandoned.

It was unfortunate that it was difficult to get reliable cost data, particularly on smaller stations. If engineers would stress the importance of more accurate records and assist in getting them, they would contribute toward the economic benefits of the users of power.

In reply to a question as to the probable dependable life of a Diesel engine, the author said that of oil engines built in the United States to the extent of 753,000 hp., engines representing less than 5000 hp. had been junked. Over 90 per cent of all the engines built in 1904, when the oil engine was in its infancy, were in operation somewhere in the world.

The useful life of a plant was really determined by obsolescence, when a more efficient type of prime mover had come to the front at a reasonable price. There would be some repairs; 1½ to 2 per cent should be figured on. The cylinders should not crack, the liners might. Bearings were like other bearings, they could be rebabbitted. If a connecting rod was bent, it could be straightened. The crankshaft might break. But over 80 per cent of the original engine consisted of frame, flywheel, and other parts not subject to breakage. He believed the life of the engine was not less than thirty or forty years.

For example, a 225-hp. engine had been built in 1907 and installed in a power plant in Texas. A public utility company bought the plant, moved the engine to another town, and operated it there day in and day out. In 1916 they put in another engine of another make and later a 500-hp. engine of the original make, the original engine being then overhauled. After overhauling, the original engine showed a fuel consumption better than that of one of the more recently designed engines and within 5 per cent of that of the other 500-hp. engine. Reliability was based on mechanical condition. With an adequate operating force, the engine should always be in good mechanical condition.

Mr. Dreyfus, resuming his discussion, said that the oil engine would not run unless it was keyed up to the highest point of efficiency. Did not that depend on whether or not the oil engine was rated at its required capacity? It was known that internal-combustion engines would limp along with poor mixtures, leaky valves, etc. and still deliver the load if they had ample capacity. But an internal-combustion engine did not have any overload capacity. Therefore, if the machine had to carry its rated capacity continuously and was not in good adjustment, it might fall out of step; but assuming that there was a little leeway in capacity, it might vary from its rated efficiency so that the management would notice the difference in cost at the end of the month.

In reply the author said that he did not believe that an oil engine ought to be loaded beyond 80 per cent of its capacity. There must be some reserve. He did not believe that an engine which would not deliver a horsepower on half a pound of oil could be operated at 80 per cent load factor. That was an increase of 20 per cent over the best fuel consumption. With a fuel variation of 20 per cent he did not believe it would carry the load.

G. C. Daniels[1] said that the Prime Movers Committee of the N.E.L.A. for the past few years had made a very interesting comparison and study of designs of small stations. About three years ago the committee had designed a 500-kw. plant; two years ago a 1500-kw. plant, and the present year a 3000-kw. plant. Various types of prime movers, such as Diesel engines, steam engines, steam turbines, had been considered. It had been found that for all load factors in plants of 500 kw. the Diesel engine was most economical; in plants of 1500 kw. the higher load factor made the turbine or steam-engine plant more efficient, and at 3000 kw. the turbine was most efficient for all load factors.

Replying to Mr. Daniels, the author said that in his opinion the N.E.L.A. report loaded the oil engine with a cost that was astonishing. The first cost was twice the installation costs in actual oil-engine plants. Such oil engines had been installed for less than $100 per kilowatt for 1500 kw. capacity. Moreover, he thought that the figures used in calculating the other costs were manufacturers' estimates rather than actual costs.

In regard to the costs, Mr. Daniels replied that they were based on plants of a reliable capacity of 500 and 1500-kw. The actual capacity was greater, of course. If three 500-kw. steam engines were installed in a 1500-kw. plant, generally two would carry the load, while two Diesels would not. The figures were based on capacity for reliable service.

L. D. Gayton[2] said that in designing a new pumping station for the city of Chicago, comparisons had been made between the economies of different prime movers—triple-expansion engines, steam turbines and Diesel engines—and in studying the Diesel, different kinds of fuel had been considered. In the case of the Diesel engine the figures used had been supplied by a number of different manufacturers. The price of the Diesel plant was something like $125 per kilowatt.

[1] Cons. Engr., 402 W. Penn. Bldg., Pittsburgh, Pa. Mem. A.S.M.E.

Fireproofing of Structural Steel

VARIOUS methods and materials for the protection of structural steel have been used. Clay tile, brick, and poured concrete are common. Clay tile does not give protection unless in turn protected. Brick is heavy, bulky, and expensive. Concrete is heavy and expensive. As a concrete incasement directly applies the heat uniformly, it is necessary to use at least a 4-in. thickness to have any real protection, for two-inches of unprotected concrete without reinforcement, poured around a steel section, does not give protection. Four to six inches at the minimum depth is expensive and under fire conditions which are of long duration is of questionable value unless it is adequately reinforced.

Plaster on metal lath has proved to be the best and most serviceable protective material. Plaster on ordinary surfaces has no fire-protective value. Ordinary adhesion is not sufficient to withstand the stresses resulting from high temperatures. When applied to metal lath a different condition exists; a mechanical bond is secured. The plaster is not only held to its work but is reinforced in every direction against the temperature stresses. The result is a tenacity and durability beyond that of any other combination of materials.—*The Iron Age*, March 5, 1925, pp. 694–695.

[1] Supvg. Engr., Hodenpyl, Hardy & Co., Peoria, Ill. Mem. A.S.M.E.

[2] Water Dept., City of Chicago, Ill.

Mechanical Springs

By JOSEPH KAYE WOOD,[1] NEW YORK, N. Y.

This paper treats the subject of mechanical springs collectively in the hope of clarifying theories of design and of assisting in the ultimate standardization of springs. After defining a mechanical spring and a mechanical-spring material, it considers the general cases of a unit cube stretched by a tensile force, of replacing the cube by a bar, and of applying the load transversely. From these are established load-deflection-rate formulas for flexure and torsion. Formulas for safe maximum load, safe maximum deflection, and safe maximum work are then derived in general terms containing constants which may be determined for stress method, material, form, etc. Under spring requirements of mechanical design, load-deflection characteristics are first considered, followed by those for safe maximum load and deflection and safe maximum work of resiliency. The paper then discusses the adaptability of springs to the requirements of mechanical design and the constants of material, dimension, stress method, and form of section, and in conclusion states that the general or collective method of treating mechanical springs should eliminate much of the complexity and diversity of the subject.

SPRINGS and their design for mechanical use have always been viewed from the standpoint of some particular type, such as helical, spiral, flat, flat leaf springs stressed flexurally, helical springs stressed torsionally, rather than from the collective or general standpoint. This may seem strange since at the beginning of the art of designing springs in 1676, Dr. Robert Hooke described to the King of England his "new discovery" of a general law applicable to mechanical springs, and at the same time demonstrated the practicability of his theory by exhibiting what is probably the first spring-driven watch recorded in history. The general law as literally stated by Hooke is that "the power of any spring is in the same proportion as the tension thereof," in which the words "power" and "tension" were undoubtedly intended to mean load and stretching, respectively.

It is the purpose of this paper to treat the subject of mechanical springs collectively, as was the natural tendency at the beginning of their use, in the hope that such a method will give a clearer picture of the greatly diversified and almost impracticable state to which the design of these very useful mechanical elements has steadily grown since Hooke's day. It is also hoped that this general method of treatment will be of considerable use in the ultimate standardization of springs.

The definitions given in connection with mechanical springs are usually vague, being based upon general elastic properties or limited to some particular geometrical form of body having elastic properties. For a general treatment of the kind proposed it naturally seems logical to base the entire structure upon a sound general definition, framed somewhat as follows:

a A mechanical spring is an elastic body whose load-deflection rate and elastic range of deflection are of values suitable for mechanical use

b A mechanical-spring material is an elastic material of the kind which when made into bodies of a shape and size suitable for use in mechanical design will function repeatedly and permanently as a mechanical spring.

Here the word "spring" has a very broad meaning, usually connected with three different types of forces, hydraulic, chemical and elastic.

The springiness connected with the deformation of a piece of material is due to an inherent "elastic" force which is generally understood to be the resultant of the electromagnetic attractions between atoms deranged from their normal space-lattice positions. The ratio of the fundamental atomic force to the relative displacement of the atoms is probably constant for a given material up to the point where slippage or plastic flow takes place. Since the resultant force and deflection are obtained by the application of simple cosine factors to the individual atomic forces and the corresponding displacements, it follows that they also must obey the straight-line law. Thus if a unit cube of metal be stretched an amount F by the tensile force P, the ratio P/F would be equal to a constant E for that material. The constant E is called "Young's modulus of elasticity" after the man who first noted this relation, and a similar constant G for a torsional force is called the "shearing modulus of elasticity." These two constants E and G form the basis of determining the suitability of a material for use in mechanical-spring design.

Considering the unit cube again in which the tensile load-deflection rate

$$P/F = E \text{ (numerically)} \quad [1]$$

if the cube length be increased to l and the area to A then

$$P/F = EA/l \quad [2]$$

and if the load P is applied transversely

$$\frac{P}{F} = E\frac{A}{l}\left(\frac{d}{l}\right)^2 k \quad [3]$$

where d is the depth of the section and k a constant for flexure. For convenience A/l might be termed the "bar index," which determines the load-deflection rate under maximum efficiency, that is, when all parts of the bar are being stressed equally. The ratio d/l might be termed the "lever index," which for flexure determines the relative values of the arms of a lever system. In a similar way k might be called the "flexure constant," which in flexure would depend solely on the form of bar section because this alone changes the character of the stress gradient.

Even when the bar of material is coiled helically or spirally this expression still holds, but the length of the bar l will not be equal to the main lever arm as shown in the lever index of Formula [3]. The more generalized formula will therefore be as follows:

$$\frac{P}{F} = E\frac{A}{l}\left(\frac{d}{D}\right)^2 k \quad [4]$$

where D = main lever arm. Reasoning similar to that above will show

$$\frac{P}{F} = G\frac{A}{l}\left(\frac{d}{D}\right)^2 k \quad [5]$$

Since torsional loading is only another mechanical method of displacing atoms, we should expect a constant relation between G and E for all metals. This very nearly happens to be the case. the small difference being due to the difference in Poisson's ratio. Assuming an average relation (about $G = 0.6\ E$), Formula [4] will apply for any method of stressing.

Continued loading of a piece of crystalline material will produce slippage or plastic flow, first, in a few of the crystals and then in a gradually increasing number until the entire material is flowing plastically. While the plastic flow is taking place locally the greater bulk of the material is yielding elastically, and this condition will be accentuated in an aggregate of mixed soft and hard crystals. It is this simultaneous occurrence of these two actions which explains the characteristic shape of the stress-strain diagram beyond the initial plastic-flow point. In continuing the tensile loading on the unit cube, a load P_m is reached at which the initial plastic flow occurs, and this, when measured with ordinary commercial accuracy, is numerically equal to the proportional limit S. With special means for obtaining considerably greater accuracy P_m might or might not be very much smaller than S, but at the same time it would approach a value equal to the fatigue endurance limit.

If the cube length be increased to l and the area to A

$$P_m = SA \quad [6]$$

and if the load be applied transversely,

[1] Consulting Engineer, New York, N. Y. Assoc-Mem. A.S.M.E.

Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, December 1 to 4, 1924, of The American Society of Mechanical Engineers. Slightly abridged.

$$P_m = SA\left(\frac{d}{l}\right)m \quad [7]$$

while the general formula for all methods of stressing, including torsional, is

$$P_m = SA\left(\frac{d}{D}\right)m \quad [8]$$

In this formula as in Formula [4] the ratio d/D is called the "lever index" and m the "stress-method and form constant."

By substituting the safe maximum load P_m in the load-deflection rate P/F, the safe maximum deflection F_m is obtained, as follows:

$$F_m = \frac{S}{E}l\left(\frac{D}{d}\right)\frac{m}{k} \quad [9]$$

or, putting $n = m/k$,

$$F_m = \frac{S}{E}l\left(\frac{D}{d}\right)n \quad [10]$$

The new elastic material constant S/E will be referred to as the "material index."

Since the safe maximum amount of work a bar of material can give is equal to the average force multiplied by the distance moved through, we may write

$$W_m = \frac{P_m}{2}F_m = \frac{S^2}{E}Vu \quad [11]$$

in which V equals the volume Al of the bar and u a constant equal to $mn/2 = m^2/2k$. S^2/E is generally known as the "modulus of resiliency."

Spring Requirements of Mechanical Design

In mechanical design it is not always necessary to have a straight-line load-deflection characteristic; in fact, in some special cases a decidedly curved characteristic with increasing load increments for equal deflection increments is desired.

In the majority of cases, however, where mechanical springs are used the natural straight-line characteristic of elastic springs is found to be suitable, particularly from the standpoint of easy calculation and control in design. It is also very convenient when making calibration charts for commercial measuring instruments. Direct proportionality will be assumed throughout the remainder of the paper.

The load-deflection rate (P/F) in mechanical design ranges from about 0.2 to 55,000 lb. per in. at the usual range of normal temperatures.

The safe maximum load P_m and deflection F_m, unlike the load-deflection rate, which for design purposes has to be within minimum and maximum limits, have only to meet a minimum requirement depending upon the service intended. As the safe maximum-deflection requirement is never very severe, being less than a few inches, the demand is always for high safe loads, the largest springs carrying 85,000 lb.

The safe maximum work or resiliency (W_m) is insufficient to gage the requirements demanded in mechanical design, because it is possible to have a very high resiliency figure with either the deflection factor too small or the load factor too small. The controlling quantities in mechanical-spring design are, first, the load-deflection rate, and second, either the safe maximum load or deflection. The resiliency has its function in spring design, however, and for that reason is considered sufficiently important for discussion. It ranges in value from 0.005 ft-lb. for small clock springs to 120,000 ft-lb. for large railway springs.

The adaptability of elastic springs to the requirements of mechanical design is the next logical step in determining the limitations which mark the mechanical-spring range.

The tensile modulus (E) of elasticity, which for the fundamental unit cube is likewise the load-deflection rate, ranges from about 6,000,000 to 32,000,000 lb. per sq. in. for the mechanically useful metals at the usual range of normal temperatures. For any one metal the modulus varies about 10 to 20 per cent for moderately high temperatures.

The proportional limit (S) as determined by the ordinary commercial testing machine ranges from about 25,000 to 270,000 lb. per sq. in. for the different spring metals at normal room temperatures. Such considerations as fatigue endurance, elevated temperatures, and unknown but severe conditions require the substitution of a lower figure in place of the proportional limit: for the first case the endurance limit, which may be considered as a "true proportional limit;" for the second case the proportional limit measured at the temperature desired; and for the third, the ordinary proportional limit modified by a factor of safety. Of course the selection of a spring material is influenced by the other physical properties, but since these do not apply directly to the calculation work they are used to influence the selection of a proper factor of safety.

The material index (S/E) or ratio of proportional limit to modulus of elasticity (using the tensile modulus), varies from about 0.001 to 0.009, while the modulus of resiliency varies from about 25 to 2500 lb. per sq. in.

To obtain high flexibility, long lengths of material have to be used, and for that reason coiling helically and spirally is done, both in flexure and torsion. Thus lengths (l) as high as 25 feet and as low as 0.250 in. are used in mechanical springs. The cross-sectional area (A) usually varies from 0.001 to 30 sq. in. The volume (V) varies from 0.0001 to 3000 cu. in.

The bar index (A/l), or ratio of area to length, varies from 0.001 to 0.10 in., while the lever index (D/d) may vary from 10 to 400.

Uniform-cross-section bars have been considered only in the previous discussion, but the introduction of a tapered bar, whether it be pointed or truncated, does not complicate the general method. In the case of a tapered spring, including the developed plan of a flat leaf spring, the bar index is very simply modified by a factor equal to the ratio of the minimum sectional area to the maximum area that is used in computing the bar index.

In the lever index (D/d) the distance D is the maximum arm of the lever system of flexural or torsional stressing. In general it might be defined as the distance from the load axis to the fixed or stationary part of the spring; for helically coiled springs stressed torsionally it is the distance from the center of the helix to the axis of the bar; for similarly coiled springs stressed flexurally it is the distance from the load axis to the center of the helix; for spirally coiled springs it is the distance from the load axis to the center of the spiral; while for flat springs stressed flexurally, it is the distance from the clamped or supported end to the free or loaded end.

The values of the stress-method and sectional-form constants for some of the important types of springs are given in Table 1.

TABLE 1 CONSTANTS USED IN GENERAL SPRING DESIGN

	k	m	n	$u \times 100$ per cent
Helical—stressed torsionally:				
Round (using modulus of elast. = E)	0.05	0.16	3.20	25
Round (using modulus of elast. = G)	0.12	0.25	2.00	25
Square (using modulus of elast. = E)	0.08	0.19	2.25	21
Square (using modulus of elast. = G)	0.20	0.29	1.45	21
Helical—stressed flexurally:				
Round and elliptical	0.06	0.12	2.0	12
Rectangular	0.08	0.16	2.0	16
Spiral—stressed flexurally:				
Round and elliptical	0.06	0.12	2.0	12
Rectangular	0.08	0.16	2.0	16
Cantilever—stressed flexurally:				
Round and elliptical	0.19	0.12	0.64	4
Rectangular	0.25	0.16	0.64	5

NOTE.—It is assumed that $G = 0.4E$ and S for torsion $= 0.63S$ for tension, which is only approximately true for materials in general.

These constants are interesting and of considerable value in selecting the right type of spring for a given service because they compare the important requirements for mechanical springs of different types when placed on an equal plane. Note that the percentage efficiency in flexural stressing of a rectangular cantilever is not as great as that in the torsional stressing of a round or rectangular bar.

As the study of fatigue of metals progresses, it is felt that the above constants will be found to bear some relation to the endurance limit.

Summary

In the preceding discussion outlining a method of general mechanical-spring design and showing the limitations of elastic springs

which are necessary to make them useful in mechanical design, four general formulas have been developed namely, [4], [7], [10], and [11]. These give:

[4] The load-deflection rate (P/F)
[7] The safe maximum load (P_m)
[10] The safe maximum deflection (F_m)
[11] The safe maximum work (W_m)

As regards the elastic constants in these formulas, it might be mentioned that in the case of torsional stressing the modulus G can be used as in the usual practice, but in this case k must be multiplied by 2.5 and m by 1.5.

The material index can be conveniently labeled in accordance with the contemplated service as follows:

1 For severe fatigue service—kinetic material index
2 For elevated temperatures—temperature material index
3 For ordinary static conditions and normal temperatures—static material index.

In the first case S should be the endurance limit, in the second case the proportional limit measured at the desired temperature, while in the third case it should be the ordinary proportional limit. In the second case E should also be slightly different, depending upon the service temperature.

In the design of a mechanical spring, Formula [4] is absolutely essential, with either Formula [7] or [10]. Formula [11], although very useful, is not sufficient for calculating a spring.

In the general or collective treatment of the subject of mechanical springs as covered above, the indications are that the complexity and diversity of this very important subject may be greatly eliminated. The very apparent relation of the component parts of each general formula to the fundamental laws laid down by Hooke and others should aid greatly in the standardization of mechanical springs. By further study and molding of this general method it will be possible to obtain a code of design which should give the mechanical designer a sense of masterly control over this part of his daily work.

Discussion

J. W. ROCKEFELLER, JR.,[1] stated that the author had made a valuable contribution to the general subject of spring design in indicating a starting point from which it might be further expanded and developed. Springs generally had been viewed from the standpoint of some particular type rather than in a general way. This fact might well be one of the chief contributing reasons for the dearth of information available on such subjects as hysteresis and temperature-modulus coefficient. While such subjects were of interest to individual users of springs, including the makers of commercial measuring instruments, it was not surprising that this class of individuals had not delved deeper into the cause of such phenomena.

The research program of the manufacturer was determined largely by practical necessity. For example, the case of direct-reading scales might be considered. Hysteresis was encountered; then by reducing the fiber stress in the steel it was found that hysteresis could be controlled and brought well within the limits of accuracy of the racks and pinions or the tapes and cams. As soon as this point was reached the attention of the manufacturer was redirected to problems of backlash, friction and theoretical pivots in which the objectionable third dimension would be eliminated. In fact, the springs themselves having been so perfected, were looked to as a possible solution for grave problems in other parts of the mechanism, and one of the most recent developments in scale design in this country or abroad had been the substitution of a flat spring for the common pivot-type fulcrum.

According to Mr. Rockefeller, the proper progress in spring research would be made only when the general importance of the subject was realized. The number of purposes for which springs were used warranted the study of springs for their own sake, if for no other reason. A fact that was given but little consideration, however, was the usefulness of a spring as a general test specimen. The laws of stress and strain that applied to springs were the general laws governing the behavior of materials. If hysteresis was a function of stress in springs, it must be present, although not easily measured, in many other forms of the same material such as the heavier types of ordnance and steel cables. It was certainly a subject of general interest.

Hysteresis, a term borrowed from the field of electromagnetics, did not seem to mean precisely the same thing to all who used it. It might be used to describe the failure of a spring, loaded for some time, always to return upon unloading to the same extended length for a given load. If used in this sense we could not make the positive assertion that the load-deflection rate had changed, since at least part of the discrepancy was due to the spring's possessing, after extended loading, a greater free length.

As to the cause of hysteresis (used in the above sense), it was quite likely to be something in the nature of internal friction. In subjecting a helical spring to a given load, there was an immediate deflection which was generally regarded as elastic. The secondary deflection taking place had usually been considered as being of a different nature from the first. The author had referred to the first as elastic and to the second as plastic. Mr. Rockefeller preferred to regard this secondary deflection as being of the same nature as the first, elastic, in fact, merely a continuance of the initial deflection, retarded greatly by internal friction in the material Perhaps this point might be clarified by analogy. It was common practice in scale design to prevent excessive vibration of the pointer in a direct-reading scale by employing a dashpot. This device would retard the action of the pointer without affecting materially the point at which it came to rest. When such a device was used in a spring scale there would be a noticeable change in the velocity of the pointer as it approached the point at which it would come to rest. This slowing down was brought about by the retarding effect of the dashpot, while the motion in its later, as well as its early phases was produced by the elastic deformation of the springs.

Fritz K. Loeffler[1] said that the paper should prove to be of considerable value in paving the way to a suitable code of design for such springs. He believed, however, that it would have been advisable to have included in the paper a formula covering the oscillations of springs. It was to be hoped that the author would consider this important item in an extension of the general method of design which he proposed for springs.

The author, Mr. Wood, in closing, said that the inclusion of the oscillation formula in a general method of spring design would be possible since such a formula expressed a simple relation between the period of frequency and the load-deflection rate. The general formula would undoubtedly contain the constant $2\pi/g$.

As to the question of hysteresis in mechanical springs, this was becoming more important every day in certain fields. The loads and deflections of springs used in sensitive length-measuring machines, aeronautical instruments, and precision weighing instruments were sufficiently small to make the errors due to hysteresis of appreciable magnitude. The hysteresis loop, formed somewhat symetrically about the true load-deflection curve, due to the lagging of the deflection on loading and of the load on unloading, was large enough, even in steel, to offer difficulties in the adjustment of these springs. The usual remedy was to give such springs a careful heat treatment, which tended to eliminate or decrease the size of the hysteresis loop.

On the other hand, although it was necessary to consider ways and means of eliminating hysteresis in springs of the type just mentioned, it was necessary to induce this phenomenon artifically in other types of springs such as in the equalization systems of electric locomotives and in the suspension systems of automobiles.

The artificial means employed were external surface friction, rubber absorbers, and liquid and gas dashpots which retarded the deflections and loads, thus forming an artificial hysteresis loop.

The purpose of having springs with a large hysteresis was to dampen or destroy undesirable oscillations. As mentioned in the paper, Mr. Wood had recently designed a flexible gear for use in electric locomotives and street cars in which the characteristics of the spring element were such as to decrease chattering slip in the drivers to a minimum.

Such questions in spring design as had been brought up during the discussion must be studied more thoroughly in order that the progress in the spring art might keep pace with that of the many other important mechanical elements.

[1] Engr., John Chatillon & Sons, New York, N. Y. Jun. A.S.M.E.

[1] Pres., Techno-Service Corpn., New York, N. Y. Mem. A.S.M.E.

Solid-Injection Oil Engines

By R. HILDEBRAND,[1] ST. LOUIS, MO.

The paper deals with the developments during the last two years in connection with a four-stroke-cycle, three-cylinder, heavy-oil engine of the solid-injection type. Combustion chambers and the effect of turbulence are discussed and the design of combustion chamber finally adopted by the author is described. The paper also describes an impact fuel pump and the spray valve developed for the engine, and concludes with some remarks on operation and the future of the solid-injection engine.

THE object of this paper is to report on the experiments and developments which the author has made during the last two years on a four-stroke-cycle, three-cylinder, heavy-oil engine of the solid-injection type. The engine has 13.5-in. by 19.5-in. cylinders, and while running at 257 r.p.m. has a normal rating of 200 b.hp., or 66.6 b.hp. per cylinder. This rating corresponds to a brake mean effective pressure of 73.6 lb. per sq. in.

The jet-injection type of engine is discussed, the injection being directly into the combustion chamber in contradistinction to injection into a "hot cup," "hot retort" or "precombustion" chamber where the fuel vaporizes and partially burns, and then enters the combustion chambers through one or a number of small holes. Furthermore, the type under consideration is for "cold starting" engines, eliminating hot bulbs, spark plugs, "punks," etc.

For the sake of brevity, the discussion will cover working principles and fundamental requirements and not details.

Combustion Chamber

There are two different types of combustion chambers used in solid-injection engines. One aims to create a violent turbulence inside of the combustion chamber, while the other does not provide for a greater turbulence than is naturally caused by charging the cylinder with fresh air.

Two examples of the first type are the Price engine represented in Fig. 1 and the Gunther engine, Fig. 2. In both cases there is a restricted passage or neck between the combustion chamber and the cylinder. The air, displaced by the piston during the compression stroke, is forced through the restricted opening, acquires a high speed, and enters the combustion chamber with a considerable velocity, thus creating a high turbulence therein. The object of this turbulence is to assist the fuel particles in meeting and uniting with the air, thus accelerating the combustion, as it is a well-known fact that fuel burns more quickly in turbulent than in stagnant air. In the Gunther engine the end of the piston is provided with an extension, or displacer, which fits loosely into the neck. The air trapped between the piston and cylinder head when the displacer enters the neck at the end of the compression stroke finds only a very restricted annular space through which it can escape. The result is that this air passes through this annular space into the combustion chamber with a high velocity, creating violent turbulence. In the Gunther engine a strong air current and an oil spray meet each other head on, causing a thorough mixing and good combustion. Three years ago the author saw an engine of this type running in the Gasmotoren-Fabrik Deutz, Germany, and was favorably impressed with its operation.

To mention a few examples of the second type of combustion chamber, in which the design does not create high turbulence, reference may be made to the Vickers engine, Fig. 3, and the Steinbecker engine, Fig. 4. In the Vickers engine the fuel enters the combustion chamber through a number of fine drilled nozzles, thus being fairly well distributed throughout the combustion chamber. In the Steinbecker engine the fuel is sprayed during the compression stroke into a small drilled connection between the cylinder and a precombustion chamber or hot retort. The air forced through this drilled hole during the compression stroke conveys the fuel into the precombustion chamber where some of it burns. This burning increases the pressure in the precombustion chamber to such an extent that the fuel is blown into the cylinder when the piston has approximately reached the upper dead-center position. About three years ago the author inspected a Steinbecker engine in Landsberg, Germany, running on coal-tar oil. The engine had a perfectly clear exhaust, and the cards shown in Fig. 5 were taken in the author's presence.

The object in mentioning the two types of combustion chambers is to show that there is a division of opinion among engineers, some of whom consider violent turbulence necessary while others do not. The fact is, and the author's experiments have proved it conclusively, that violent turbulence is only necessary in connection with certain designs, which will be discussed later. However, if the combustion chamber, fuel pump, and spray nozzles are correctly designed and function properly together, engines employing only a moderate turbulence will operate satisfactorily. This is also proved by Vickers, Steinbecker, M.A.N., Deutz (new vertical engine), Leissner and others prominent in the Diesel-engine field, who do not employ violent turbulence within the combustion chamber, but find the turbulence caused by charging the cylinder with fresh air sufficient.

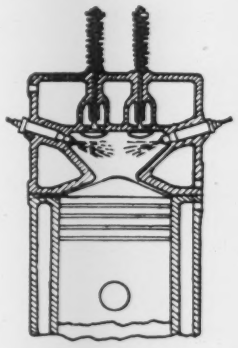

Fig. 1 Combustion Chamber of Price Engine

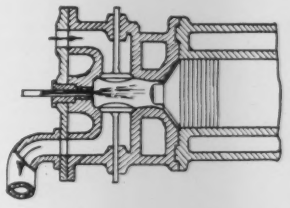

Fig. 2 Combustion Chamber of Gunther Engine

Fig. 5 Indicator Card from Steinbecker Engine

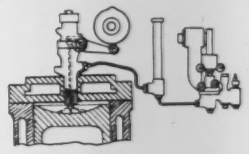

Fig. 3 McKechnie Engine (Vickers and Maxim)

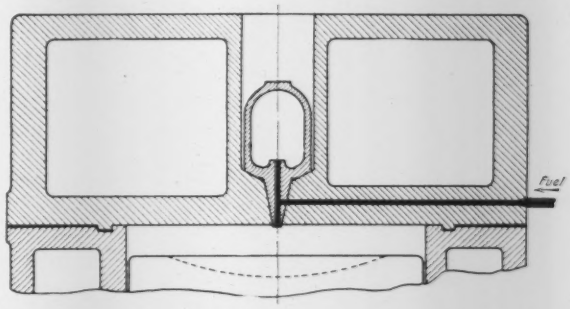

Fig. 4 Steinbecker Engine Diagrammatically Shown

Violent turbulence within the combustion chamber is objectionable from a thermodynamic point of view and should be avoided as much as possible, as it causes undue heat losses by radiation. These heat losses can be revealed by an indicator card. If a card is taken after shutting off the fuel supply to the cylinder, it will show a negative work area because the compression line will lie above the expansion line. If no heat were lost during the cycle, the compression and expansion lines would coincide. The negative area of the indicator card shows the amount of heat lost by radiation, expressed in work. Figs. 6 and 7 are cards taken of combustion chambers shown in Figs. 1 and 9, respectively. Reference to the combustion chamber of Fig. 9 will be made later. These

[1] Chief Engineer, Diesel Division, Fulton Iron Works. Mem. A.S.M.E.

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two combustion chambers differ fundamentally in so far as the one in Fig. 1 falls under the group employing violent turbulence, while in the one in Fig. 9 there is only moderate turbulence. The cards shown are taken from cylinders of equal size, while the engines ran at the same speed in both cases. They speak for themselves and prove that excessive turbulence is objectionable.

When the author began to design the solid-injection engine referred to above, he decided not to imitate the combustion chambers shown in Figs. 1 and 2 which were at that time attracting the most

Fig. 6 Fig. 7

FIGS. 6 AND 7 INDICATOR CARDS WITH FUEL SUPPLY SHUT OFF
Fig. 6: From combustion chamber of Fig. 1
Fig. 7: From combustion chamber of Fig. 9

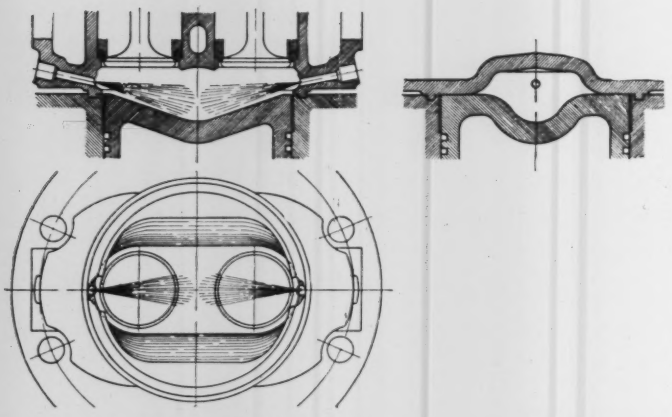

FIG. 8 COMBUSTION CHAMBER WITH OPPOSING SPRAY NOZZLES IN LINE

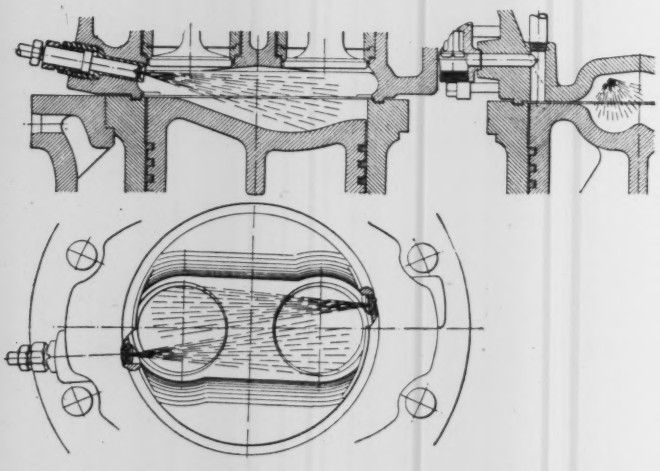

FIG. 9 COMBUSTION CHAMBER WITH OPPOSING NOZZLES OFFSET

attention. He did not like the idea of separating the combustion chamber from the cylinder by a restricted opening or neck, as such a combustion chamber is not sufficiently compact, having too much water-cooled surface for its volume. He finally decided to use only the space between the piston and the cylinder head as the combustion chamber, and to make it as compact and with as little cooling surface as possible. Fig. 8 shows the combustion chamber with which the experiments were started. Notice that two opposing nozzles were employed, and that the fuel spray did not fill the air space. The compression reached 440 lb. and was later reduced to 350 lb. per sq. in. The highest brake mean effective pressure obtainable with clear exhaust was about 35 lb. per sq. in., while the engine is supposed to have a brake m.e.p. equal to 73.6 lb. per sq. in. and a reasonable overload. The rated power was not reached on account of the peculiar arrangement of the spray nozzles.

The two sprays in the combustion chamber shown in Fig. 8 opposed each other, i.e., their axes were in line. When the engine was originally designed, there was some doubt as to whether the opposing nozzles would form the best arrangement. At that time this arrangement appeared to be correct, because the prevailing theory was, and in general is still, that a spray would not penetrate more than a few inches through a combustion chamber filled with dense air, and that in an engine of the type shown in Fig. 1 the sprays would reach barely to the center of the combustion chamber. Experiments have demonstrated that these theories are wrong and that the fuel actually penetrates through the whole combustion chamber, provided it is not consumed or burned before reaching the opposite walls. For this and other reasons to be mentioned the arrangement of two opposing nozzles should be avoided in this engine.

What actually happened in the combustion chamber shown in Fig. 8 was that the two opposing sprays penetrated each other. This resulted in an excessive fuel supply at the center of the combustion chamber, a supply which increased with the load. With violent turbulence in the middle of the combustion chamber, this excess fuel at the center will be dispersed by the air current and a clear exhaust can be obtained. However, without this violent turbulence the central part of the combustion chamber becomes oversaturated with fuel due to opposing sprays, and the engine is bound to show a smoky exhaust. This explains why the nozzles in the engine shown in Fig. 1 employing violent turbulence may oppose each other and yet give satisfactory results. It also explains why the combustion chamber shown in Fig. 8, employing only moderate turbulence, resulted in smoky exhaust even at moderate loads. Therefore, if the designer arranges the nozzles to oppose each other, he must provide for violent turbulence at the center of the combustion chamber. However, as stated before, violent turbulence is undesirable from the thermodynamic point of view, and therefore it is best not to arrange the nozzles to oppose each other.

FIG. 10 PISTON HEAD OF FIG. 9

Realizing these facts, the nozzles were offset and the piston top changed accordingly,

FIG. 11 CYLINDER HEAD OF FIG. 9

as shown in Fig. 9. It will be noticed how completely the fuel spray fills the air space. The piston tops were provided with two offset cavities in line with the nozzles, flaring out in opposite directions. The piston top and the bottom view of the cylinder head are shown in Figs. 10 and 11, respectively. This design was chosen so that the two sprays could not meet, to give the combustion chamber the same shape as the spray, and to make the spray length as great as possible. The combustion chamber shown in Fig. 9 gave an excellent result, and it was possible to carry full load (brake m.e.p. 73.6 lb. per sq. in.) and 40 per cent overload with perfectly clear exhaust. The engine carried 50 per cent overload with a slightly smoky exhaust.

Several other combustion chambers similar to the one shown in

Fig. 9 were tested. They deviated from the design shown in so far as the nozzles were arranged at different angles. The experiments with these combustion chambers proved conclusively that the sprays should not come in contact with the hot piston top or other hot surfaces. For the sake of comparison, several cylinder heads of the high-turbulence type were made and tested. The combustion chambers employing moderate turbulence proved to be the best. No favorable tests were given by the high-turbulence chambers.

The combustion chamber which was finally adopted and used in the Hildebrand design of solid-injection engine, shown in Fig. 9, has the following characteristics:

a Compactness with small cooling surfaces, the piston clearance being the combustion chamber

b Moderate turbulence resulting in comparatively small heat losses by radiation. An explosion-like combustion is avoided with moderate turbulence and consequently the engine runs quietly and starts without heavy detonation. Moderate turbulence also results in a fairly progressive combustion which approximates the Diesel or constant-pressure cycle. Further it develops greater mean effective pressures (at a predetermined maximum initial pressure) than is obtainable with high turbulence, which causes an explosion-like combustion characteristic of the Otto cycle

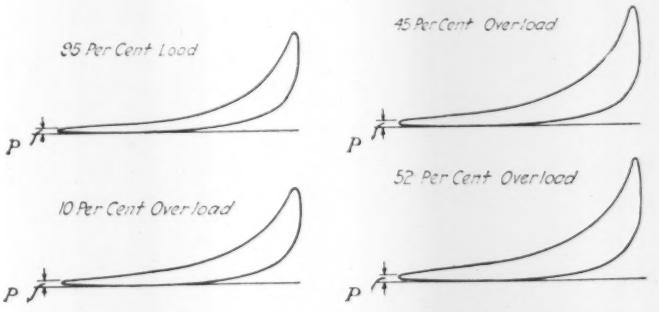

FIG. 12 INDICATOR CARDS FROM THE HILDEBRAND ENGINE

c The nozzles are offset, preventing the sprays from meeting and penetrating each other

d A long spray length is used, and consequently comparatively large spray nozzles are permissible

e The combustion chamber consists substantially of two long cones similar in shape to the sprays. This makes it possible for the fuel to penetrate nearly the whole volume of air contained in the combustion chamber

f Clear exhaust and low fuel consumption are obtained at high brake mean effective pressure

g The walls of the cylinder head and the piston tops exposed to the heat are curved and are consequently flexible, which minimizes the stresses therein.

The cards obtained from the solid-injection engine of the Hildebrand design of the mentioned size and speed and involving the above features are shown in Fig. 12. These sharp-pointed cards indicate an expansion almost to the atmosphere and consequently a low fuel consumption.

FUEL PUMP

When designing a fuel pump for an engine of the solid-injection type, which sprays the fuel directly into the combustion chamber, it must be remembered that a nozzle gives a well-atomized fuel spray only as long as the fuel has sufficient speed. The spray becomes coarse and dense as soon as this speed drops below a certain velocity, and delayed vaporization and combustion manifested by smoky exhaust will result. Since the velocity depends on the pressure in the nozzle, it is obvious that this pressure should never drop below a certain limit while the nozzle is spraying. For this reason the nozzle pressure must rise suddenly when the plunger starts to lift, and drop suddenly when the effective pump stroke ends, i.e., when the by-pass valve opens. This is first of all a problem in kinematics; but there are other considerations which should not be overlooked, and which will be taken up later.

From the above it is evident that a sluggish plunger motion at the beginning of the fuel injection is undesirable for solid-injection engines. This holds good particularly for those types with only moderate turbulence in the combustion chamber. The conditions are different in engines having violent turbulence. In the latter the spray may be coarser and yet the engines may give clear exhaust, as here the strong air current (turbulence) disperses the coarsely atomized fuel.

A cam-operated pump has a sluggish plunger lift. The author began his experiments with such a pump, and Fig. 13 shows its characteristics. The cams used were of the tangential type having $^5/_{16}$ in. lift. Steeper cams could not be used as they caused excessive hammering and wear on the cams, cam rollers and roller pins.

Referring to the chart it will be noticed that the plunger velocity starts from zero and increases steadily to a maximum. This steadily increasing plunger velocity—characteristic of all cam-operated pumps—is highly undesirable as it results in a very uneven nozzle velocity. The nozzle orifice cannot be selected to suit

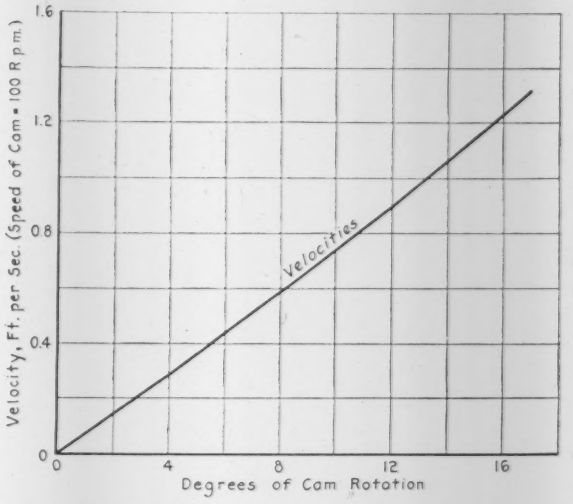

FIG. 13 PLUNGER VELOCITY OF CAM-OPERATED FUEL PUMP

the mean plunger speed but must be small enough to secure a sufficient nozzle speed at the beginning when the plunger moves slowly, so that the fuel at the beginning of the spray may be sufficiently atomized. Thus a cam-operated plunger pump requires a small nozzle orifice. The small nozzle opening, being selected with reference to the beginning of the plunger movement, is inadequate when the plunger acquires a higher speed for the remaining part of the fuel-injection period, because it gives too fine an atomization and consequently an insufficient penetration of the fuel. The finer the atomization, the less the penetration of the fuel.

This explains why the combustion chamber shown in Fig. 9, employing only moderate turbulence, required nozzle openings of 0.022 in., or smaller, when used in connection with a cam-operated fuel pump. Even then the engine could barely carry its rated load as indicated by the smoky exhaust. These small nozzle openings not only limited the engine's capacity but also caused too high a pressure in the fuel pump, leaky plungers, and much trouble with the cams, cam rollers, and the helical gears driving the pump. This also explains why cam-operated fuel pumps work fairly satisfactorily in connection with combustion chambers employing violent turbulence. Here larger nozzle openings can be used because a coarseness of the spray at the beginning of fuel injection is less harmful, as previously explained.

From the experience with the fuel pump just described, it appeared necessary to use either combustion chambers employing violent turbulence, or a pump with approximately constant plunger velocity during the period of fuel injection. It also appeared desirable to use hardened and ground steel plungers and liners and to drive the pump directly from the crankshaft without the use of gears. Therefore the author designed a new fuel pump driven directly from the crankshaft. Its arrangement relative to the engine can be seen from Fig. 14. The unit shown comprises the fuel pump itself, the speed governor, the overspeed or safety governor, and the centralized engine control, with an arrangement

to by-pass every second fuel discharge as the pump driven from the crankshaft works on the two-stroke principle while the engine is of the four-cycle type.

Each plunger of this new pump, which is of the impact type, is driven by an individual eccentric mounted on an extension of the crankshaft. Shortly before the crosshead of the eccentric reaches its central position it strikes the plunger. The latter acquires a high velocity almost instantaneously, as seen from the chart Fig. 15, which shows the characteristic of this pump. A

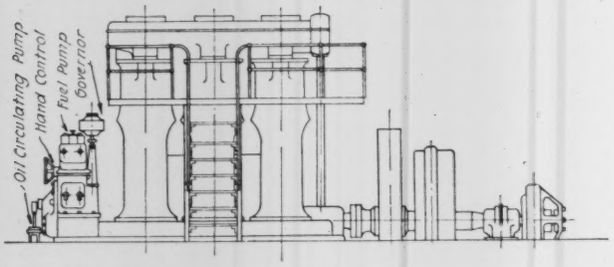

FIG. 14 HILDEBRAND ENGINE, SHOWING LOCATION OF FUEL PUMP

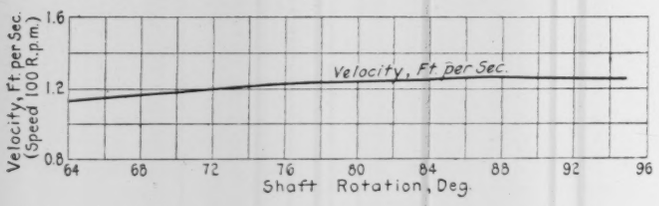

FIG. 15 PLUNGER VELOCITY OF IMPACT FUEL PUMP

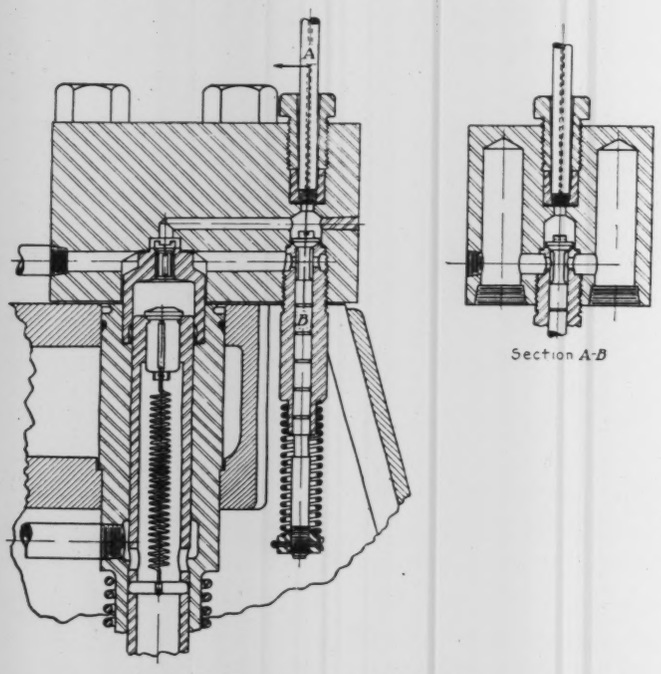

FIG. 16 DETAIL OF IMPACT FUEL PUMP

well-atomized spray is insured from the beginning of the fuel injection by this sudden and high plunger speed. On the chart, Fig. 15, it will be noticed that the impact pump has nearly constant plunger velocity which gives a nearly constant pressure in the nozzle during the effective plunger stroke. This made it possible to obtain a uniform and well-atomized spray during the whole period of fuel injection with a nozzle having an orifice of $^1/_{32}$ in. and larger. These large nozzles furnished not only well-atomized sprays but also great spray lengths without requiring excessive pressure in the pump. Thus the impact pump made it possible to use large nozzles in connection with combustion chambers employing only moderate turbulence to obtain clear exhaust up to full load (73.6 lb. per sq. in. brake m.e.p.) and 40 per cent overload. Even 50 per cent overload could be carried as above mentioned.

The impact pump described has given full satisfaction. It is very strongly built and the bearing areas of the eccentrics and wristpins are as large as the space permits. The noise which the impact makes is not loud, and is less than the noise of a cam-operated pump. This is due to the fact that in the impact pump the only mass to be suddenly accelerated is a light hollow steel plunger and its spring. In a cam-operated pump the roller, roller pin, crosshead, plunger, and its spring are simultaneously accelerated. In the latter case the masses are great and the acceleration comparatively small, while in the former case the mass is small but the acceleration is high.

FUEL-OIL VOLUME

Fuel oil is an elastic substance; it compresses when subjected to pressure and it reëxpands when the pressure is released. Therefore it is obvious that the total volume of fuel oil subject to compression and reëxpansion, i.e., the volume measured from the suction valve of the pump to the orifice of the nozzle, should be as small as possible. The smaller this volume the more quickly will the oil be compressed to spray pressure, and the more quickly will this pressure be relieved when the by-pass valve opens at the end of the injection period.

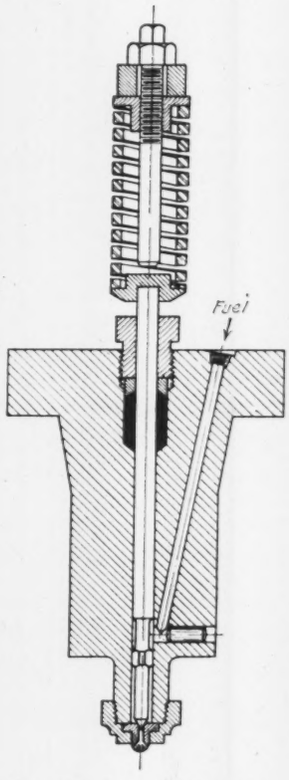

FIG. 17 SPRAY VALVE OPERATED WITHOUT VALVE GEAR

A slow pressure rise at the beginning of the injection is highly undesirable as pointed out above. For the same reasons a slow pressure drop at the end of the fuel injection period should be avoided. From this it appears that the larger the oil volume contained in the fuel lines, the coarser will be the spray at the beginning and at the end of the injection. This coarseness of the spray is not readily noticeable with combustion chambers employing violent turbulence, because, as mentioned above, a strong air current (turbulence) disperses the coarsely atomized fuel. The conditions are different in combustion chambers employing only moderate turbulence. Here the author could plainly notice the harmful effect of a large oil volume in the pump and fuel pipes. Accordingly, he built the impact pump mentioned above with as small a pump volume as possible, and provided air domes in the pump head next to the by-pass valve. Fig. 16 shows a part of the plunger, plunger liner, and the pump head with suction, discharge, and by-pass valves.

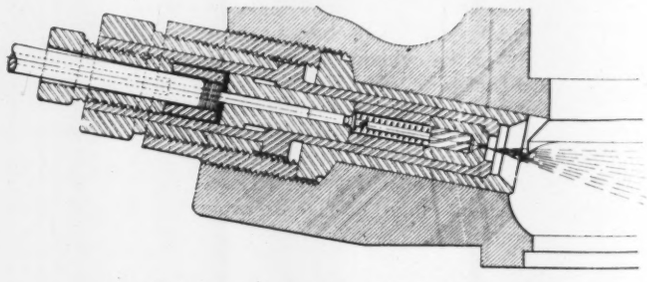

FIG. 18 SPRAY VALVE OF HILDEBRAND ENGINE

It will be noticed that the hollow pump plunger serves as the suction chamber, and that the suction valve is riding on the upper plunger end. This reduces to a minimum the oil volume in the pump. The air domes serve the same purpose as those provided at the discharge side of an ordinary plunger pump. These domes, large by-pass valves, and wide and short returns pasages to the

suction basin allow the oil to escape readily when the by-pass opens. The result of this entire arrangement is a well-atomized spray, giving good combustion without employing violent turbulence in the combustion chamber.

Another fundamental requirement which must be fulfilled to obtain good combustion with solid-injection engines employing only a moderate turbulence is that the fuel oil in the fuel line shall reëxpand freely toward the by-pass valve when the latter opens. If the oil cannot reëxpand toward the by-pass valve, it will expand toward the nozzle and will cause after-dripping. For this reason it is undesirable to install a check valve in the pump head preventing the reëxpansion of the oil contained in the fuel pipe, when the by-pass valve opens. However, a check valve in the fuel line may be used with success if placed at the nozzle or as close to it as possible, for instance, in the tee where the fuel line branches off to the two nozzles (provided the combustion chamber is equipped with two nozzles).

Spray Valve

Three types of spray valves may be distinguished: One employs a needle operated by a valve gear closing toward the combustion chamber, as shown in Fig. 3. The second type employs a needle closing toward the combustion chamber without using a valve gear as shown in Fig. 17. Here the needle end is reduced in diameter. The fuel pressure acting on the differential area of the needle, lifts it. The third type eliminates the needle and valve gear, and employs nothing but a spring-loaded check valve closing toward the pump as shown in Fig. 18. This is the one used with good results in the Hildebrand design of the solid-injection engine. This spray valve, or nozzle, has one central orifice and a fluted plug which imparts to the fuel a whirling motion.

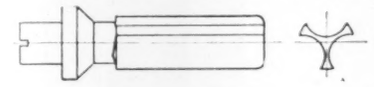

Fig. 19 Check Valve First Used

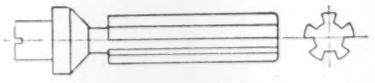

Fig. 20 Check Valve Finally Adopted

The only difficulty experienced at the beginning with the nozzle shown in Fig. 18 was to make and to keep the check valve tight. If this check valve is not tight, gases from the combustion chamber enter the interior of the nozzle through its orifice when the by-pass valve in the pump opens, i.e., when the pressure in the pump and fuel pipe is released. The gases which thus enter the nozzle will prevent a sudden rise in pressure behind the nozzle orifice at the beginning of fuel injection and will cause a predripping at the nozzle and consequently a smoky exhaust. This predripping at the nozzle affects equally engines using moderate turbulence and those employing violent turbulence in the combustion chamber.

The difficulty with the leaky check valves was eliminated by improving their guides. Fig. 19 shows a check valve greatly enlarged with a three-pronged guide which centered well when the valve rested as shown in the upper half of the illustration; but when the valve rested as shown in the lower half of the illustration, it failed to center properly. To improve the centering of the check valve, a guide having a greater number of flutes was employed as illustrated in Fig. 20. There is no difficulty in keeping a check valve of this design tight, provided dirt is kept out of the fuel oil.

Light springs on the check valves have given better results than heavy ones. The plug referred to above, which imparts a whirling motion to the fuel, improves the spray and the combustion. Fig. 21 shows the spray of the nozzle illustrated in Fig. 18. The pump used to obtain this spray is hand operated for testing.

Operation

The solid-injection engine—provided it is well designed—can be operated successfully if properly attended to. There are, however, certain requirements which the operating engineer must fulfill in order to obtain good results.

First of all, the fuel oil has to be strained and filtered, and provision must be made that absolutely no dirt can reach the fuel after it has been filtered. This can be accomplished if the strainers and filter are properly designed and if the operator is careful. Secondly, the operator should test the nozzle by a hand testing pump before he places it in the engine, to be certain that the check valve is tight. After the designs were finally perfected, the author failed to find any operating difficulties either with combustion chambers employing only moderate, or with those having violent, turbulence.

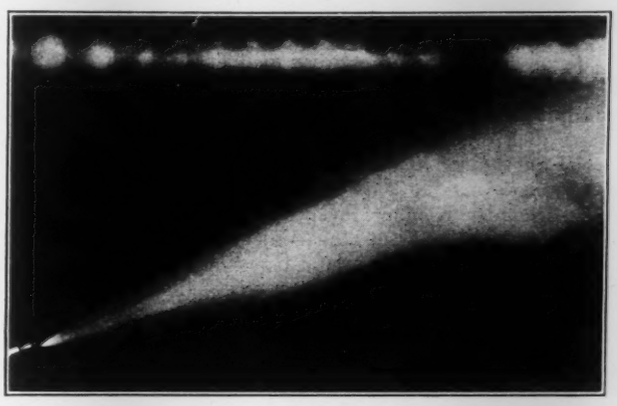

Fig. 21 Spray from Nozzle of Fig. 18

Lubrication of Diesel Engines

USUALLY, three different kinds of lubricants are used: One for the power cylinder; one for the crankcase system, lubricating the crankshaft bearings, crankpins and wristpins; and one for the compressor. For the power cylinders and the crankcase pure mineral oils, uncompounded with animal or vegetable oils, and neutral filtered as a final process of manufacture, should be used. The lubricants for the compressor are either pure mineral oils or mineral oils compounded with vegetable oils. All oils used should stand the following emulsion test: Mix one ounce of oil and one ounce of distilled water, heat to 180 deg. fahr., and then shake violently for five minutes, after which the mixture should stand and separate completely during one hour.

Nothing but a well-suited lubricant for the power cylinders will prove successful. Unsuitable oils will cause carbon deposits and sticky piston rings, resulting in blowing pistons and consequently contamination of the crankcase oil with carbon. Some years ago I visited a Diesel power plant where the bad effects of improper cylinder lubrication were strikingly evident. A hot crankpin box caused a shutdown, and an inspection revealed a carbon deposit about $^3/_8$ in. thick all over the crankcase. The plant had run out of Diesel cylinder oil and had used ordinary red engine oil instead, which formed carbon, caused the rings to gum up, the piston to blow, and thus fouling the crankcase oil caused a hot box.

It is a wrong idea to think that an inferior oil copiously applied to the cylinders will do as well as a good oil. In fact it is poor policy to feed more lubricating oil, even of the best grade, to the power cylinder than is absolutely necessary. Too much oil will gum up the piston rings and will cause oil to drip from the liners. This oil having been subjected to heat, is contaminated by carbon, which thus finds its way into the crankcase. This should be avoided as much as possible.

If the liner is free from scale and well cooled, and if a suitable lubricating oil is used, about 3 to 4 drops per feed per minute should be enough for each cylinder. Two-stroke-cycle engines need a little more oil, as a part of the lubricant finds its way into the exhaust ports and is thus lost. Some manufacturers of four-stroke-cycle engines do not provide for any special lubrication for the power cylinders, but depend entirely on the splashing in the crankcase to lubricate the pistons and liners. This splashing is considerable in a high-speed engine equipped with a force-feed oiling system.—R. Hildebrand in a paper published in *Power*, Feb. 17, 1925, p. 255.

Aerial Surveying Equipment—Helicopters—Production Airplanes of Metal

Discussion of Papers on These Subjects at the A.S.M.E. Annual Meeting, December, 1924

AT THE recent Annual Meeting of the A.S.M.E., three papers devoted to aeronautics were read at a session held under the auspices of the Aeronautic Division of the Society, Dr. Sanford A. Moss,[1] Chairman of the Division, presiding. Ernest Robinson[2] delivered an illustrated address on Equipment Used for Aerial Surveying[3] in which he described equipment developed since the war for making maps mathematically to scale from photographs; Alexander Klemin[4] reviewed the aerodynamic and construction data available on the subject of the helicopter;[5] and Edmund Burke Carns[6] presented a paper on Production Airplanes of Metal.[7] A summary of the discussion of these papers is given below.

Discussion of Paper on Aerial-Surveying Equipment

In discussing Mr. Robinson's paper, Archibald Black[8] asked if, with aerial cameras in their present stage of development, it would not be possible to devise a radio-controlled airplane and camera for mapping enemy territory at low altitudes without risking lives of pilots or photographers.

Mr. Robinson answered that it was not anticipating the possibilities too much to expect that, but that there was still a long way to go.

Mr. Black then asked what would be the approximate cost per mile of aerial survey made for the purpose of locating airplane routes, that was, for mapping a strip of territory perhaps several hundred miles long and a few miles wide. New York to Chicago might be regarded as a typical location for such a survey.

Mr. Robinson said that it would depend entirely upon the scale of the map. For reconnaissance purposes, for example, it would cost in the neighborhood of $25 per running mile, at a scale of approximately 1200 to 2000 ft. to the inch.

To Mr. Black's question as to the lowest altitude of flight that would give satisfactory negatives for stereoscopic purposes, Mr. Robinson replied that it would depend upon weather conditions. For stereoscopic purposes it was considered desirable to fly as low as possible, say, anywhere from 1000 to 2000 ft., as the lower one flew, the greater would be the resulting angle of displacement.

Mr. Black also asked how the camera was prevented from oscillating in its gimbal mount. Mr. Robinson explained that the camera remained steady due to the slight friction between the bearings, and that, at the present time, it was held level by the operator. With the new gyroscopic control and the rectifying unit that would probably be finished in about three months, the camera would be held by a worm and worm drive, and would synchronize with the gyroscope within a variation of about 0.3 deg. The complete new equipment would weigh about 90 lb. The present one weighed about 50 lb.

R. Beaver[9] stated that the U. S. Geological Survey was at present making a map of the United States by quadrangles having an area of about 15 by 20 miles each, and that it took three surveyors two season's work, approximately, to map one quadrangle. It would then seem that the Geological Survey would be the one to profit mostly by an aerial survey.

Mr. Robinson said that the Geological Survey had been depending upon the assistance that the Air Service was able to give both with the Bagley camera and the K-3, and that no doubt this assistance would greatly increase in the future and result in very appreciable savings, especially in cases where the ordinary methods were presenting more than usual difficulty.

H. W. Crozier[1] said that he had seen the work done by the Air Service in the determination of a large amount of blown-down timber in the Pacific Northwest, on the occasion of a great storm or cyclone some three or four years ago. The United States Air Service ran five strips across this country and from the data these provided it was possible to make a very accurate determination of the condition of the standing timber in various parts and make a fairly accurate estimate of just how much timber was standing in certain particular localities.

Mr. Robinson made the statement that up to now the stereoscope had not been used as an aid in plotting contours, but only for reconnaissance purposes—in order to show the hills and dales and the local conditions, rather than to determine dimensions. He believed that there was a machine nearly completed, constructed by Mr. Brock of Philadelphia, that would plot contours not less than 20 ft. apart, and that some very interesting experiments had been conducted with his device.

Discussion of Paper on the Helicopter

In discussing Professor Klemin's paper, Henry A. Berliner,[2] said that he had read it with considerable interest and had fortunately been able to discuss some of its main points with the author in Dayton last October.

Broadly speaking, he agreed with Professor Klemin's conclusion, but would like to give a few reasons why in his case he had concentrated on the helicopter-airplane type rather than on the pure helicopter.

It was Mr. Berliner's belief that any helicopter which possessed propellers large enough to effect a safe landing with a dead engine would, due to the enormous resistance of the propellers in horizontal flight, have insufficient speed to make it a practical machine. Moreover a safety device of this sort did not take care of a failure of the driving mechanism or control mechanism of the machine.

Also, in all pure helicopters built today, the large amount of driving mechanism had increased the weight to such an extent that the available useful load had been as small or smaller than in his case. This statement might seem ambiguous in view of the fact that the De Bothezat helicopter and the Oehmichen had both lifted large useful loads a few feet into the air. But it was well known to those who had made static propeller tests that a surface directly in the propeller slipstream might enormously increase the thrust of that propeller. This effect did not become negligible until the surface (or in the case of full-size machines, the ground) was distant more than two diameters from the propeller. In fact, as far as Mr. Berliner knew, no full-size controllable helicopter had definitely risen above the region of ground effect. In this category he included his old machine.

Professor Klemin had objected to the helicopter-airplane on the ground that it would not be efficient in vertical climb. This statement Mr. Berliner believed was erroneous. The upper limit of the rate of climb of all helicopters was of course the horsepower of the engine multiplied by the propeller efficiency and divided by the weight of the machine. This of course would give an exceedingly high figure. In an individual case the rate of climb might be arrived at in another manner. Suppose the propeller thrust were 2000 lb. and the weight of the machine 1600 lb. The machine would then climb until the 400 lb. available thrust for climb was balanced by the vertical resistance of the structure, assuming a

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[1] Engr., Mech. Research Dept., General Electric Co., River Works, West Lynn, Mass. Mem. A.S.M.E.

[2] V.-P., Fairchild Aerial Camera Corp., New York, N. Y. Mem. A.S.M.E.

[3] Published in Mechanical Engineering, March, 1925, p. 170.

[4] Assoc. Prof. Aeronautics, in Charge of Aerodynamic Lab., New York University, University Heights, N. Y. Assoc-Mem. A.S.M.E.

[5] Published in Mid-November issue of Mechanical Engineering, 1924, p. 739.

[6] Jamaica, L. I., N. Y.

[7] Published in Mid-November issue of Mechanical Engineering, 1924, p. 733.

[8] Consulting Aero Engineer, Garden City, L. I. Mem. A.S.M.E.

[9] Works Mgr., Rochester Industries, Inc., Rochester, N. Y. Mem. A.S.M.E.

[1] Engr. Sanderson & Porter, San Francisco, Calif. Mem. A.S.M.E.

[2] Designer of the Berliner helicopter, 2841 Tilden St., Washington, D. C.

constant thrust, which was probably correct at these speeds. Even in the case of a helicopter-airplane with a large projected area this should permit a climb well in excess of the 600-ft-per-min. climb predicted by Professor Klemin for his hypothetical machine.

Taking it all in all, Mr. Berliner believed that the helicopter-airplane had the better chance of success, particularly in view of its comparative simplicity. Both the Oehmichen and the De Bothezat machines had experienced considerable mechanical difficulties in spite of the careful workmanship that was put at least on the De Bothezat.

Max M. Munk[1] found that Professor Klemin had omitted to mention the difficulty of a vertical flight near the ground due to the wind and other disturbance of the atmosphere. He thought, further, that it did not follow from one test that tandem airscrews were aerodynamically superior to a pair placed side by side. He could not see how and why theoretical considerations indicated this generally.

Dr. Munk also called the author's attention to the fact that the disk represented an aerodynamic problem very different from that of a windmill. The air passed the windmill circle but not an actual disk, and no reliance should therefore be placed on an analogy between a windmill and a disk. Further, no rotative motion could directly change the momentum parallel to the axis, but had only a bearing on the horsepower and blade action.

It would be interesting to note that at tests made in the laboratories of the National Advisory Committee for Aeronautics, drag coefficients somewhat higher than 0.6 had been obtained. These tests had not as yet been published.

Archibald Black[2] expressed the indebtedness he felt for Professor Klemin's valuable contribution to the subject. He stated that helicopter development to date had been very disappointing, and that accomplishments so far had not been in any proportion to the time and thought given to the problem. At the same time, he thought that those who had followed these activities were more or less convinced that a practical design could be developed. The question looming largest in his mind was, What were we going to do with it if we got it? Nothing would be gained by developing something without utility. The field of the helicopter in his mind seemed to be limited to that of replacing kite balloons, and who could tell if kite balloons would continue to be a military necessity in the future? He doubted if any one had yet suggested a really commercial application for the helicopter.

This condition would discourage many practical men, and the more practical they were the more it would discourage them, from giving attention to helicopters. He thought that the proponents of this type of aircraft would serve their cause best if they could tell what it could be used for. Some really practical suggestions for its application would act as an incentive to bring more minds to bear on the problem. These suggestions would also furnish something concrete to strive for. Before preparations were made to spend a tidy sum on development there should be some promise of at least getting back what was to be put in. It was very nice to talk of working for the glory of it, but the glory in helicopter development had already been spread out so thin that there was little substance left. Before we went any further, therefore, we should ask what chance there was of getting our money back.

Edward P. Warner[3] said that it seemed to him that the fact that no commercial application for the helicopter was apparent in the present stage of development was no reason for abandoning attempts to develop it. The application would doubtless come when the machine had been carried to a considerably higher point technically.

As to the more technical parts of the paper, there was, first, the subject of wind-tunnel work. He thought Professor Klemin had done the more scientific of the helicopter investigators a little injustice in that respect. Not much of the wind-tunnel work had been published, to be sure, but a very considerable amount of it had been done. In fact the M.I.T. in the past two years had been carrying out investigations for two helicopter designers and as much work had been done as it seemed feasible to do in the wind tunnel at the present time.

[1] National Advising Committee for Aeronautics, Washington, D. C.
[2] Cons. Engr., 25 Brixton Rd., Garden City, L. I., N. Y.
[3] Assoc. Prof. Aeronautical Engrg., Mass. Inst. of Tech. Mem. A.S.M.E.

Of course it was true of most helicopter inventors, as it was of most of the early workers in the field of the airplane, that they did not make use of the laboratory, but as the Wright brothers had made use of it, so had those who were most likely to achieve success with the helicopter. Furthermore, he thought that direct and special experiments were not so necessary at present because a large amount of the necessary data had already been collected.

When airplane design was started nothing was available; nothing was known about wings, nothing about propellers. The helicopter, however, used a propeller depending upon the same principles as the airplane propeller, and a helicopter design could be based very largely on work already done in testing propellers for airplane service.

On tandem propellers Professor Warner wanted to add his own comment to Dr. Munk's. He also could not see why the tandem propeller was likely to be more efficient than the other, and he was disinclined to believe that it would be so on the basis of experiments made in 1910. Recent tests at the M.I.T. had also given negative results.

On ground effect, the point raised by Mr. Berliner was of tremendous importance. It had generally been underestimated. The lift of the helicopter could be easily increased 50 per cent if close enough to the ground, and, as Mr. Berliner had said, he thought there was no evidence that any helicopter had been wholly clear of the ground effect. Furthermore, there was another point which he did not mention, and that was that the ground effect was very helpful to stability. A helicopter could be used for what the airplane pilot called "contour chasing" very successfully, because in flying across undulating ground the machine would keep at a constant distance above the surface. If it sank even a few inches, the propeller thrust would increase and it would rise again. It had, therefore, stability of vertical motion, which it would not have at a considerable altitude. Furthermore, the effect of a dihedral, the value of which when flying at high altitudes was rather doubtful, certainly far less than the same disposition of the wings of an airplane, was much increased when near the ground, because as the machine rolled, one propeller was shooting the air off sideways, so to speak, while the one that had been depressed was closer to the ground and was thrusting the air directly against the ground and therefore got an increased lift. The lateral stability was very much greater near the ground than that at a high altitude. Stability and control were the most serious problems, and he saw no way in which we could go very far in the investigation of that problem in the wind tunnel.

Finally, there remained the performance of the helicopter to be considered. Professor Warner was in very close agreement in general with what Professor Klemin had said about the horizontal speed, although he thought that perhaps he should not be quite as optimistic as he was. He certainly was not as optimistic about the usefulness of the propeller turning as a windmill in an oblique glide.

He had not seen the Dutch figures to which Professor Klemin made reference, but he had not been able to get any conformation on ordinary propellers without feathering blades, and he was inclined to think there would be little if any gain in using an oblique instead of vertical descent.

Professor Warner thought that the performance was going to be very much better than the author had intimated for a vertical climb. If a curve of power required to fly an airplane was plotted against the rate of climb at a given air speed, the curve was a straight line. It took a certain amount of power to maintain flight, and the power required to climb was directly in proportion to the rate of climb. For the helicopter that rate went upward very sharply. By using a propeller of a little higher pitch than was common, a condition could be secured in which it would actually take less power to climb than to hover. Some airplane propellers gave that.

Professor Warner had made some calculations from typical airplane propellers which might be adapted to helicopter work, and had found that a propeller which gave a thrust of 15 lb. per hp. when hovering, would give a little more than that when climbing 200 ft. per min., and at 3000 ft. per min. would give a thrust of over $7\frac{1}{2}$ lb. per hp., or more than half as great again as when hovering.

Once we got away from ground effect, the problem of climb

seemed to be a very easy one. Even a small excess of thrust was enough to send the rate of climb up to 600 or 1000 ft. per min. or perhaps considerably more than that.

W. P. Suplee[1] stated that it might be interesting to recall the fact that the original work of the Wright brothers was with the helicopter. They had one of these toy machines that was spun between the hands. They played with it, and then began to try to improve it.

In that connection, attention might be called to the fact that an airplane could be mathematically considered as an element of a propeller blade spinning on a vertical axis with an extremely large radius; so that inversely it might be possible for us to apply some of our information regarding the airplane to solve the helicopter problem.

Wm. J. Hammer[2] said that there were several types of helicopters that might have been brought out in connection with the paper. Wilbur Kimball, of the Aeronautical Society, in 1909 tested out a helicopter machine of a very unusual type—which might have been discussed by Professor Klemin to advantage—in which he used twenty propellers driven by wire transmission. The total surface represented something like 320 sq. ft., and the entire supporting and circling frames with the propellers weighed only 112 lb. It was driven by a very small engine, about 50 hp. Mr. Kimball had considerable difficulty not only with the engine but with the transmission also. The reason for the interest lay in the fact that it would probably not be possible to build a propeller of very large size with a sufficiently strong sustaining enclosure to represent an efficiency such as was secured by these multiple propellers.

Of course there were disadvantages of interference and mechanical disadvantages which were self-evident, but personally the discusser had always been a great believer in the helicopter.

Wm. F. Durand[3] said that he believed Professor Klemin's paper to be a distinct contribution to the literature of the helicopter. He had been very much interested in considering the possibilities of the helicopter as a scientific or engineering achievement, and as a practical and useful addition to our means of air transport.

It had already been demonstrated that it was possible to rise from the ground. Therefore the possibility of the helicopter had already been carried forward to the point of complete demonstration. The possibility of what could be done with it, just how useful it could be made, was not so clear.

During the War, in 1917, as at that time Chairman of the National Advisory Committee for Aeronautics, Dr. Durand had favored the stimulation of effort and energy in all directions which pointed toward possible useful results, and among these of course the helicopter naturally came in for consideration. As one means of stimulating such effort he remembered the proposition had been brought forward that a prize should be offered. He had drawn up at that time the conditions which should be fulfilled by the competitors in such a prize competition. A number of aeronautic firms were written to regarding the proposition, but the Committee at that time had no adequate appropriation directly available for this particular purpose, and had to go to the War Department which refused the money, and so the prize competition died. However, the attitude of the War Department at that time should not be too seriously criticized. The creation of a practical, useful helicopter for war purposes was not a matter of a few weeks or a few months, but more probably of several years, and they did not believe, and Dr. Durand was not prepared to state that he believed himself, that such a machine could be developed in time to prove of any very great value in that particular war.

The developments since those days had justified this view. Of course, the intensive efforts had not been carried forward, but such efforts as had been made in Government plants in this and other countries had shown that the development of a practicable machine was beset with many, many difficulties. It was not too much to hope that many of these difficulties would be surmounted, but they had not been as yet.

And then there always remained the question of just what we would do with it and just in what way it would be distinctly superior to existing types. So far as military purposes were concerned, it would have of course the distinct advantage of getting out of a small back yard and of coming down in the same area, and furthermore, perhaps the most important advantage from a military standpoint, of hovering, at least in quiet air or in reasonably uniform wind, pretty nearly stationary over a given point, and so taking the place of the observation balloon—with, however, the ability of getting away quickly in case of the appearance of enemy airplanes. That would be a very distinct advantage, and military authorities of course were perfectly ready to agree to the points urged in its favor.

But aside from these, it was hard to see any other very great advantage, and of course the operation of the helicopter was hedged about with a certain number of other difficulties. It labored under the difficulty of realizing the two necessary functions of sustentation and propulsion, either by two separate sets of organs or at least by one set of organs, and either one function or the other must be carried out at a rather low efficiency. If our sustentation failed, how were we to realize the necessary slowness of descent in order to provide for safety? But these were difficulties which were perhaps incidental, and he would certainly be very far from saying or even believing that the next two years might not perhaps bring about some combination of structural elements which would furnish a satisfactory type of machine of this form, and that it would take its place as a useful element in our means for air transport.

Professor Klemin, answering Mr. Berliner's objection to the De Bothezat helicopter without planes, said that the reason for the De Bothezat helicopter's being so complicated was due to the fact that vertical descent was intended. If, on the other hand, we made a concession and said that we would not descend vertically with the motors out of commission but descend on the glide, it would be possible to use much smaller diameters of propeller for lifting. Also in the De Bothezat helicopter four lifting screws had been used, and that number could be diminished. Therefore a pure helicopter could be designed which would be much less complicated than the De Bothezat machine and which would, he still thought, show at least results equal to those obtained with the airplane-helicopter of Mr. Berliner.

Regarding Dr. Munk's question as to the superior efficiency in lift of two tandem airscrews, the quickest way of answering that would be to consider the lower blades as throwing the air upward with a certain velocity. Therefore the upper screw of the combination acted as if it were in downward motion, and an airscrew acting in an upward wind would provide more thrust than an airscrew acting in still air.

Regarding stability investigations, which Professor Warner thought could not be investigated in a tunnel, Professor Klemin thought it was merely a matter of patience and ingenuity in devising the experiments, and that a great deal could be learned in stability by methods analogous to those employed in studying the stability of an airplane.

He also thought that the same answer could be made to the respective merits of climbing vertically and climbing on the glide as would be done with a helicopter-airplane. That point, he believed, had already been considered in the paper. When climbing vertically the entire energy was employed in overcoming gravity. Also there was only one system of transformation of energy, and the parasite resistances were comparatively small. On the inclined climb there would be a much greater velocity to increase the parasite resistance, and also there would be two transformations of energy, one in the propeller and the other in the sustaining system. In the helicopter pure and simple there was only one transformation of energy, which of course might be taken to tend to greater efficiency.

Regarding a remark made by Mr. Freeman on the Magnus effect, there was still another point to be brought up, and that was that if by means of the Magnus effect sufficient difference of pressure could be produced, there might be a much greater drag than in the case of a wing section. In other words, adequate lift could be obtained very readily but there might be less efficiency than when the lift was obtained by movement of an inclined plane, as in the case of an airplane.

Mr. Black's suggestion that we could do all that we needed with

[1] Ch. Engr., Constr. Operation, Thos. Wolstenholme Sons & Co., Frankford Ave. & Westmoreland St., Philadelphia, Pa. Assoc-Mem. A.S.M.E.
[2] Past-president of the Aeronautical Society, New York, N. Y.
[3] Prof. M.E., Stanford University, Cal. Pres. A.S.M.E.

the airplane at present was a very valuable contribution to the subject, but it seemed to Professor Klemin that it would be very fascinating not only to be able to land on a restricted spot but also to be able to hover over it.

Mr. Hammer was far too modest, as he had done more in the initial stages of the airplane than was generally realized. He had brought out the point of multiple propellers. That was a subject that Professor Klemin had touched briefly in the paper. It seemed to him that another line of attack, brought out by Mr. Hammer, might in a modified form lead to some very valuable results, namely, the consideration of a multiple power plant in which we were no longer concerned with the troublesome problem of descent with the motors out of commission.

Mr. Hammer suggested twenty propellers. Perhaps twenty propellers was a little too many, but we could imagine a system in which we would have six motors of moderate power with six propellers. Then those propellers would be designed solely for lifting purposes and all problems of descent with motors out of commission would be disregarded; therefore it might be possible to design propellers of comparatively small diameter.

Another advantage of the multiple-power system and the multiple-propeller system was that the propellers could be kept down to the minimum weight. Theoretically a propeller could be built which, with 1 hp., would lift 10,000 lb., but to lift 10,000 lb. with 1 hp. the propeller would weigh so much that the device could not conceivably get off the ground. It seemed to Professor Klemin, therefore, that inventors and designers might consider the possibilities of a multiple-power, multiple-propeller system as one having certain advantages.

Discussion of Paper on Production Airplanes of Metal

In discussing Mr. Carns' paper, H. C. Richardson[1] stated that the production of airplanes of metal, as suggested, appeared feasible to him, but that there was some doubt in his mind whether the methods proposed would lead to weight economy in this type of construction.

The various devices proposed were direct and ingenious and undoubtedly would in quantity reduce costs of fabrication and assembly.

The use of felt washers would reduce shock, abrasion, noise, and the nicety of workmanship required in the type of construction proposed, but, he feared, would violate the requirements of determinate structures. Also in service, it was to be feared, the felt would become compacted and provide the very slackness it was intended to avoid. By giving a force a chance to move, it was given a chance to do work. To avoid corrosion, particularly at sea, the washers would require impregnation with oil or paraffin.

Referring to details, he would point out that in Fig. 2 the bolt head should be turned at right angles to avoid secondary flexure of struts, as the wing beams deflected under load.

The next discusser, Charles B. Kirkham[2] stated that the outline of types of construction and estimates of cost of metal airplanes seemed to be based on many assumptions that were open to argument.

The assumption that the biplane would be best suited for economical construction had not been proven by the author and was contrary to Mr. Kirkham's experience. Furthermore, as the real purpose of airplanes for commercial purposes should be to promote speed of transportation, the biplane, especially in the larger sizes, would be very much inferior to the internally braced monoplane in speed, carrying capacity, or cost of production, if made in anything like production quantities.

Mr. Carns' scheme of wing construction using tubular spars was new only in details. The use of tubular spars was poor construction from the standpoint of weight as the principal loads on the wing spars produced bending stresses, for which tubes did not have the most economical section.

The fuselage construction shown seemed to be the usual type of wire-trussed construction except as to the use of metal and the form of contour formers used, and in Mr. Kirkham's opinion would be far from being as cheap to produce or of as light weight, for a given strength, as a welded steel tubular construction such as was now in general use by many of the most progressive designers. The use of duralumin covering for the fuselage would certainly not be as cheap as the usual fabric, and in his opinion would be less desirable as it would be more easily damaged, especially if used in flat shapes such as shown by Mr. Carns' design.

Mr. Carns made the bare statement of cost of tools, overhead, production, and design costs for a machine to carry 1800 lb. useful load, and as these figures were so definite, it would be of great interest if he would give more information as to their source.

For instance, what size airplane was it contemplated would be required to carry this 1800-lb. load with a Liberty motor, and what would be its weight loaded? What would its performance be?

What percentage of the total weight of plane would be of duralumin?

What amount had been allowed for "inspecting and testing every bit of material used in the airplane" as mentioned on page 736?

How many turnbuckles and wire ends had to be made up for this airplane and at how much each?

What was the basis for the statement of the cost of building the last ten planes?

What size wheels were to be used and what would be their cost?

Charles T. Porter,[1] in discussing Mr. Carns' paper, stated that the author had submitted figures on the production of a type of metal airplane that were interesting, and possibly attainable, but in getting these low figures he had made sacrifices in design which were very serious, so serious, in fact, that the discusser doubted whether the resulting product would be worth producing at all.

The airplane, with resistances of 200 to 300 lb. per ton, was at best an inefficient means of transport, and to compete at all it must be as efficient as the limitations of the art would permit.

The performance of an airplane was dependent on its pounds per square foot and on its pounds per horsepower, the common element in both being weight. This all-important factor was one which Mr. Carns had seen fit to ignore. Without reliable unit weights for ships of different sizes, the design and its production possibilities were of little interest.

It was stated in the paper that the design might be employed for airplanes varying from small sport planes to ships of 1800 hp. This statement could not go unchallenged. The design as shown might be feasible for small sport ships, but for large airplanes it would be impossible. The large airplane was only feasible by the use of the utmost refinement of design. It must be as nearly as possible a "One-Hoss Shay," and every detail must have a balance of design that was entirely lacking in the product under discussion.

Time did not admit of a vigorous analysis of all the details, but a brief discussion of the principal structural members would show the point at issue.

1 The wing beams carried the total weight of the airplane in transverse bending, in a vertical plane, and in addition end loads imposed by the wing truss. The resulting stresses due to combined bending and flexure, while complicated, were possible of mathematical analysis and showed that in general the moment of inertia in the vertical plane should be from 20 to 40 times that in the horizontal plane. Obviously a steel tube did not meet these requirements. The discusser had recently designed a steel beam that successfully passed Army Air Service tests, which consisted of two 2-in. steel tubes forming the top and bottom chords of a trussed beam 17 in. deep. A single tube for this beam would have been utterly impossible.

2 The author gave a detail of the beam fitting at the point of intersection with the strut. This location was generally the point of maximum bending moment, and yet the design showed a vertical bolt cutting the tube at its region of maximum stress. There was also a very serious eccentric pull that would probably cause failure.

3 In the design of the body the reliance on solder to take the chord increments of the diagonal shear members would be open to serious objection, especially in large airplanes where all members were loaded to maximum capacity. The method of building the body in three parts was a good production proposition and one that was used successfully in building welded-steel fuselages, but the method of joining the parts was not so good. The aluminum plug had in itself small shearing resistance, hence it was supposed

[1] Commander, U.S.N. Navy Dept., Bureau of Aeronautics, Washington, D. C.

[2] Pres., Kirkham Products Corp., Garden City, L. I.

[1] Ch. Engr., Huff, Daland & Co., Ogdensburg, N. Y. Mem. A.S.M.E.

that reliance was placed on the steel collar. If so, why the plug? Again, the use of a single bolt to develop the strength of the tube was questionable. Either the thickness of the tube must be increased by brazing in a collar, or sufficient bolts must be used to give bearing area enough to develop the required strength. As a joint the proposed design would have a low efficiency.

4 The cost figures were also open to challenge. The author showed that after the first two machines, sixteen men could build ten airplanes in six weeks, which meant 422 hours per airplane. This meant a labor cost of about one-half the cost of material. There was nothing in the design of the airplane to indicate such figures unless most of the parts were machine-made, in which case a large capital cost would be necessary for dies, punches, and presses. Present-day costs indicated a ratio of labor to material of about three to one. Material reduction in this ratio could only be obtained by standardization and crystallization of design. In the present state of rapid development such a condition was not desirable, and when it came, the product must show greater refinement of design than that proposed by the author.

In the general discussion from the floor, Mr. Miller[1] stated that Professor Warner, who had been compelled to leave the meeting a little earlier, had left a note regarding some points that he wished to be brought up.

First he desired to commend the author on the use of felt to eliminate vibration.

Another point brought up by Professor Warner and on which Mr. Miller also agreed with him, was that it was still an open question whether the built-up fuselage, consisting of tubular structure with fittings, was as economical as the welded-steel fuselage except in large-quantity production.

Another interesting point that Professor Warner desired to have brought up was the fact that Rohrbach in Germany had found that there was a considerable amount of electrolytic action between steel and duralumin when in contact.

Mr. Miller did not agree with the author that the type of wing construction he used would be economical as regarded weight or economical in cost of construction, even in large or small quantities, except possibly on certain types of machines. It might work out on the particular type he was using, but certainly not on very large machines.

Archibald Black said that he did not intend to enter into a detailed discussion of Mr. Carns' design, because there were one or two points on which he disagreed with him both as regarded weight and economical construction, and he was afraid that it would take too much time to thresh out each one of these piece by piece.

Ever since he had become interested in the operation of airplanes, anything that was concerned with cost always caught his eye, and the one thing that he had noticed first in the paper was the manufacturing-cost figures. These were surprisingly low, but he thought they might be within the realm of possibility.

In considering airplane-manufacturing costs, one was inclined to compare the author's figures with the present-day cost of manufacturing airplanes for the Government, but he felt that the comparison would be very misleading.

First of all, Government planes were usually built to order in small numbers, such as ten to fifty. It was very seldom that a manufacturer got an order for more than fifty. That would imply that the manufacturer was allowed to build fifty airplanes exactly the same, but he did not think there was any case where this has been done. Almost invariably the manufacturer was asked to make changes from time to time while the designs were going through production. Any one familiar with production would realize what that meant.

On top of that inefficiency, Mr. Black quoted the case of one of the large airplane companies which charged 250 per cent for overhead plus 15 per cent for profit plus 5 per cent for contingencies, and yet at the end of the year closed with a deficit in its financial report. Such figures, therefore, would be very misleading for purposes of comparison.

A. F. Johnson,[2] said that the author had been very patient with some of the criticisms, and that it was not entirely certain that all of the criticism was merited in view of the fact that he seemed to admit that he was merely presenting something to talk about. It seemed to Mr. Johnson that the airplane would ultimately, and very shortly, come to the status where it could be made in quantities. Such an airplane, of course would not be of the large type. It would undoubtedly be something similar to the small two-seater, low-powered machine which was now being utilized in Europe and in this country.

When we asked a man to submit specific costs on a basis of production in a field where no production had ever existed, we were more or less asking something that was manifestly impossible. There was, however, no doubt that the use of standard fittings, while possibly not applicable to the entire plane, would be applicable to a large percentage of the structure of the plane, and that those fittings, being essentially small, and being standardized, through perhaps conferences of the aeronautical engineers, which would determine what was good and what was not good, would render possible something of the nature of the progress that had been achieved in the automotive field.

[1] Curtiss Aero & Motor Corporation, Garden City, L. I., N. Y.

[2] Fleischmann Co., New York, N. Y. Mem. A.S.M.E.

New Wood-Treating Process

DATA on a one-movement process for impregnating timber with zinc chloride and petroleum. While zinc chloride is a good preservative for wood under certain conditions, its use has been decreasing because of certain difficulties that it produces, especially when the timber is used in hot and dry places. To overcome these handicaps it was proposed to use zinc chloride with petroleum, but this requires a double treatment with or without an intervening period of seasoning, and a double treatment has certain obvious objections.

In the present development this was eliminated by using a mixed zinc chloride and petroleum-oil preservative in the form of an emulsion. The emulsion is prepared by mixing asphaltic residuum and distillate fuel oils in proportions to give a suitable viscosity. This mixture is agitated with a certain proportion of zinc chloride solution (in proportions given in the original article), then pumped under high pressure (about 2000 lb. per sq. in.) and at a temperature of about 170 deg. fahr. through an emulsifying valve. This valve so disperses the zinc chloride solution that an emulsion of sufficient fineness and stability for the required purpose is obtained. No artificial stabilizer is necessary. Certain practical suggestions as to the conduct of the process are given in the original article.

From tests carried out it would appear that there is a tendency for the preferential absorption of zinc chloride by the wood, so that, for example, if an emulsion containing 5 per cent of zinc chloride when used for a treatment giving a retention of 10 lb. per cu. ft., the zinc chloride absorbed would not be 5 per cent of 10 lb., or 0.5 lb., but some figure higher than this amount. It would further appear that this excess absorption of zinc chloride is constant for a given kind of timber and increases with an increase in difference between gross and net absorption. A mathematical solution of the problem of equilibrium concentration of zinc chloride is given in the original article. On the other hand, it appears that the zinc chloride is adequately distributed in the oil-treated area.

Semi-industrial treatments have been in progress at Somerville, Texas, for several months. This work is being done in coöperation with the Atchison, Topeka, and Sante Fe Railway and the Grasselli Chemical Company. It is anticipated that these treatments, involving 3500 ties and started during August, 1924, will be nearing completion by Feb. 1, 1925. (A. M. Howald, Senior Industrial Fellow, Mellon Inst. of Research, University of Pittsburgh, in *Railway Review*, vol. 76, no. 7, Feb. 14, 1925, pp. 304–310, illustrated, *d*)

A new gasoline feed pump operating by the action of brass bellows expanded by an electromagnet energized from the battery and called the autopulse is made by the Ireland & Matthews Mfg. Co., Detroit, Mich. This bellows action also comprises a helical spring which supplies the energy necessary to deflate the bellows, thus delivering the fuel drawn into it during expansion. Illustrated details of the construction of the pump are given in *Automotive Industries*, vol. 51, no. 21, Nov. 20, 1924, pp. 890–891.

Operating Experiences in Pulverized Fuel

AT THE joint meeting of the Cleveland Engineering Society, and the Cleveland Sections of The American Society of Heating and Ventilating Engineers and The American Society of Mechanical Engineers held in Cleveland, Ohio, on January 13, 1925, John Wolff presented a paper on Pulverized Coal in Large Boilers, which was published in the January, 1925, issue of MECHANICAL ENGINEERING. In connection with this paper P. W. Thompson and E. H. Tenney presented extended discussions on operating experiences with the Detroit Edison Trenton Channel plant and the Cahokia Station of the United Electric Light & Power Co. These two discussions follow directly:

Pulverized-Coal Burning at the Detroit Edison Company's Trenton Channel Plant

By PAUL W. THOMPSON,[1] DETROIT, MICH.

THE Trenton Channel plant of The Detroit Edison Company is located on the Trenton Channel branch of the Detroit River about 18 miles south of Detroit and at the southern end of the 120,000-volt transmission line which passes around the northern limits of the Detroit metropolitan district, terminating at the Marysville power house, about 50 miles north of Detroit.

Pulverized coal is burned at the Trenton Channel plant. After considering the possible arrangements and location for the coal-preparation house, it was decided to build a separate house for this purpose just east of the boiler room. This decision was arrived at primarily for two reasons. First, the three-boiler-battery arrangement did not lend itself well to locating the preparation plant inside the boiler house, and second, because of the difficulty in obtaining sufficient green-coal storage space.

All coal is received by rail and is broken to a size not exceeding 1¼ in. in a Bradford breaker having an hourly capacity of 315 tons. Before distributing to the green-coal storage bunker, the coal passes over a distributing belt having a magnetic pulley which removes all small pieces of iron not removed in the breaker.

From the storage bunker the coal passes downward through steam heated driers where the moisture is reduced to less than 2 per cent. The driers are so arranged that with reasonably dry coal to start with a certain amount of drying is obtained by merely inducing air through the driers without using any steam. For this purpose warm, dry air is drawn from the economizer gallery. During that portion of the year when the coal contains considerable moisture, more intensive drying is obtained by the use of steam which is bled from the turbines driving the plant auxiliaries.

Eight boilers are installed having a working pressure of 415 lb. gage, with a total temperature of 700 to 725 deg. fahr. The boilers each have a total heating surface of 29,087 sq. ft. and the furnace volume is 25,140 cu. ft. The furnace bottom is cooled by a tube screen having a surface of 1220 sq. ft. This tube screen is novel, in that it is fed separately from the boiler and the water-circulating system is entirely separate. The steam generated in the tube screen, however, discharges into the steam drum of the main boiler. In case of a failure of a tube of the tube screen, resulting in the loss of its water, the safety of the boiler proper would not be jeopardized.

There have been no screen-tube failures except during the first few months of operation when a large amount of raw make-up was added to the system. Owing to the high rate of evaporation from the screen tubes (about 30 lb. per sq. ft. per hr.), scale formed in a short time, which resulted in a few bagged tubes. However, since a raw make-up has been eliminated, there has been no further trouble.

Cottrell precipitators have been installed in the breechings leading to the stacks in order to prevent an excessive amount of ash from escaping. The application of this equipment to power-house use is new and it is yet too early to predict just how satisfactory it will be. Preliminary tests, however, indicate that with the precipitators the amount of ash discharged from the stacks is less than with stoker-fired boilers having no cinder-catching devices.

Some trouble was experienced with the walls of the first boiler, and it was found necessary to rebuild the two front walls. The trouble was due to operating the boiler for long periods of time at low ratings which resulted in the failure of the steel supporting beams holding the inner lining of the furnace. At low ratings little air is required for combustion, although the furnace temperature is comparatively high. Since the air required for combustion is used to cool the supporting members of the furnace lining, when operating at low ratings this quantity was insufficient for this purpose. The method of supporting the inner lining has now been changed so that even at low boiler ratings no trouble is expected.

The Trenton Channel plant first went into operation with one 50,000-kw. generating unit about six months ago. Since then two additional generating units of the same size have been added.

Very little trouble has been experienced in connection with the coal preparation or the burning of pulverized fuel. Such difficulties as have been experienced are minor ones and few of them are attributable directly to the use of pulverized fuel. The plant operating men are very enthusiastic over the simplicity of the boiler-house operation and the ease with which the boilers are controlled.

In general the experiences to date indicate that the burning of pulverized fuel at the Trenton Channel plant is working out very satisfactorily and that the choice of this method of burning fuel was a wise one.

[1] Ch. Asst. Engr., Power Plants, Detroit Edison Co. Assoc-Mem. A.S.M.E.

Pulverized Coal at Cahokia

By E. H. TENNEY,[1] ST. LOUIS, MO.

ONE OF the most interesting and important characteristics of the use of fuel in pulverized form is the fact that fuels of widely different quality may be successfully used.

The fuel used in the Cleveland district is what we farther west regard as a high grade of bituminous coal, in that it contains low percentages of moisture, ash, iron, and sulphur, and is of high heating value. The characteristics of this coal tend to higher boiler efficiency and greater ease of furnace operation than is the case with the low-grade Illinois coals used in the St. Louis district. Some of the fuel now being pulverized and burned at Cahokia could not be burned satisfactorily on a stoker. In discussing the characteristics of Cahokia Station, St. Louis, therefore, as compared to other stations, this difference in the quality of fuel should be taken into account with relation to the differences in station design and in arrangement of the fuel-preparation and burning equipment.

It is interesting to note the effect of the various non-combustible elements in a low-grade coal upon the economy of a station using pulverized coal.

Moisture. High moisture results in an increase in power for pulverizing, in greater quantities of air for transporting, and in less effective burning. The coal used at Lake Shore contains from 1¾ to 3 per cent moisture as received and hence does not require drying. At Cahokia the coal contains 12 per cent moisture and must be dried down to at least 5 per cent before satisfactory milling and transporting can be accomplished. This drying is accomplished by the use of waste stack gas. About 25 per cent of the gases at 410 deg. fahr. are drawn through the coal as it passes downward through the driers to the mills. The temperature of the gases is reduced to 140 deg. fahr. and the moisture in the coal to about 5 per cent.

Ash and Iron. The coal at Cahokia contains a high percentage of ash, frequently in excess of 16 per cent. This ash is of a quality which is largely accountable for one of our most difficult furnace problems, namely, the erosion of side walls. The ash contains 18.5 per cent of iron and its fusing temperature is 2000 deg. fahr. The coal contains 5 per cent sulphur. These particular chemical elements when brought in contact with the refractories of the side

[1] Ch. Engr. Power Plants, Union Electric Light & Power Co. Mem. A.S.M.E.

walls at high temperatures and high rates of combustion seem to be just what is needed for melting the surface of the firebrick. In other words, the ash becomes a flux for the firebrick. In the first eight boilers at Cahokia the refractory is of a quality sometimes referred to as "balanced brick." That is, it contains about equal parts of alumina and silica as the chief ingredients. It is possible partially to overcome this side-wall erosion by changing the quality of the brick used, and experiments to determine just what the proper ingredients making up such a brick should be are now being made. Other steps also have been, or are being, taken for bettering this condition: The shape of burners has been changed to throw the body of the burning fuel away from the side walls; air has been admitted along the side wall in a way to form a protective blanket against the high furnace temperature; the use of the burners immediately adjacent to the side walls has been discontinued; side walls have been built vertical instead of sloping inwardly as on the earlier installation. Water-cooled side walls are now being installed. One furnace with the Murray fin-type water-cooled wall is now being erected, and two with tubes embedded half-way in the brickwork will be built in the spring. At Ashley Street station we are now building a furnace for pulverized fuel, with Bailey water-cooled sides and bottom.

The use of very low-grade coal at Cahokia has, of course, introduced other factors in the operation of the furnaces which in plants using higher-grade coal are of minor importance. Chief of these are the greater quantities of ash to be handled throughout the boiler as well as in the bottom of the furnace. The low fusion temperature of the ash results in slag on boiler tubes and in the ashpit, and requires much care in the keeping of water surfaces clean. Where erosion of the side walls takes place this molten brick finds its way to the ashpit and frequently requires special attention in its proper removal.

The sluicing of ashes is another distinctive feature of Cahokia. The accumulation of dust from boilers and furnaces is dropped through doors in the bottom of the ashpits into a sluiceway through which water is jetted at high velocity. This mixture of ash and water flows to a receiving pit, from which it is pumped to the adjacent property.

Table 1 is a report of a test which was run on one of the 1800-hp. boilers at Cahokia, operating at 214 per cent of rating, and which was made prior to the changing over to the very low-grade coal now being used. The boiler efficiency is shown to be 84.5 per cent.

TABLE 1 RESULTS OF BOILER TEST AT CAHOKIA POWER PLANT, UNION ELECTRIC LIGHT AND POWER COMPANY

(Babcock & Wilcox cross-drum boiler, 18,010 sq. ft. of heating surface)

Duration, hr.	21.18	Pressures, lb. per sq. in., abs.	
Per cent rating	214	Superheater	331
Coal as fired		Drafts, in. of water	
Moisture, per cent	6.82	Furnace	0.18
Ash, per cent	11.06	Uptake	0.80
B.t.u. per lb.	11,713	Pressure feeder air	13.5
Fired per hour, lb.	13,161		
Per cu. ft. of combustion space per hr., lb.	1.12	Analysis of gases, uptake, per cent	
		CO_2	14.7
Ash, per cent:		O_2	4.3
Comb. in furnace ash	..	CO	0
Comb. in flue dust	2.8		
Water, lb.		Heat account in per cent	
Evaporated per hour	107,114	Heat absorbed by boiler and superheater	84.5
Evaporated per lb. of coal	8.18	Loss in dry gases	10.5
B.t.u. per lb. of steam	1210.6	Loss in water vapor	5.0
Temperatures, deg. fahr.		Loss in incomplete comb.	0.3
Air to furnace	80	Radiation, error, and unaccounted for	−0.3
Gases leaving boilers	527		
Feedwater	198	Total	100.0
Superheated steam	718		

TABLE 2 CAHOKIA STATION OPERATING RESULTS

1924	B.t.u. per lb. coal	Boiler effy., per cent	B.t.u. per kw-hr. output	Thermal effy., per cent	River water Temp., deg. fahr.	Back pressure, in hg.	Avg. Sta. water rate
Aug.	11,016	79.95	17,951	19.02	79	1.593	12.00
Sept.	10,787	81.38	17,403	19.62	69	1.243	11.88
Oct.	10,783	80.26	17,224	19.99	62	1.148	11.59
Nov.	10,793	80.06	17,290	19.75	49	1.052	11.78
Dec.	10,771	80.22	17,093	19.98	35	0.798	11.63

COAL HANDLING AND PREPARATION

	Kw-hr. per ton
Power required for:	
Unloading, conveying and crushing	0.40
Drier fans	2.66
Pulverizing mills	16.15
Air compressors (for transporting pulv. coal)	8.52
Total power for coal handling and preparation	27.73

Cost of handling and preparation including maintenance = 36 cents per ton.

Cahokia station is not equipped with economizers, the reason being that with the cheap fuel available the saving to be effected was not sufficient to warrant the additional investment cost. With economizers as a part of Cahokia's boiler equipment 6 per cent would probably be gained in boiler efficiency, and a corresponding lesser B.t.u. per kw-hr. output would result. Some such adjustment must be made when comparing Cahokia economy with that of other plants equipped with economizers.

Table 2 gives the operating results for the last five months of the year 1924.

The Storage and Handling of Fuel Oil in Industrial Plants

UNDER the auspices of the Materials Handling Division a session on oil handling and storing was held at the recent Annual Meeting of the A.S.M.E. at which C. G. Sheffield[1] and H. H. Fleming[2] presented a paper entitled The Storage and Handling of Fuel Oil in Industrial Plants. This paper discussed the subject with reference to land oil-burning installations and dealt with such practical problems as location, foundations, steel tank construction, handling, and heating. J. A. Shepard,[3] member of the Executive Committee, Materials Handling Division, acted as chairman of the meeting. The paper was printed in full in the Mid-November, 1924, issue of MECHANICAL ENGINEERING, pp. 771–773. The discussion follows.

Harold T. Moore[4] submitted a written discussion in which he said that under "Construction of Steel Tankage" the authors pointed out the minimum thickness of plates that should be used, namely, $^3/_{16}$ in. for the shell and $^1/_8$ in. for roof, even for small tanks. Industrial-plant owners were not so likely to take liberties with safety factors as the oil producers, who more frequently purchased tanks for temporary use. Some of the tanks installed in oil fields where producing wells were declining were built with as low a factor of safety as $1^1/_2$. Permanent tanks installed at refineries or industrial plants, where reasonably long life was anticipated, should have sufficient metal in the plates and joints to insure a safety factor of not less than $2^1/_2$.

Corrosion might occur on the external side of the tank bottom from sulphurous acid if cinder fill was below the sand cushion, or from humus acids in the soil. The top ring and inside of the roof might likewise experience deterioration from moisture and from sulphurous or hydrocarbon gases condensing on the shell. Ample thickness of plate with periodic painting was therefore justified from the standpoint of insurance as well as for safety reasons and reduced fire hazard.

The use of wood roofs with tar-paper covering on steel shells in order to economize on initial cost was poor economy. Combination steel and wood tanks could not be kept tight and therefore increased the fire risk as well as the insurance premium over that of all-steel tanks.

Self-supporting globe roofs, while slightly more expensive than conical-top tanks, especially in the larger sizes, wrote Mr. Moore, were more attractive in appearance than the standard-type tanks. This design, which had a crown radius equal to the tank diameter, was built in sizes up to 10,000 bbl. capacity, i.e., 54 ft. diameter by 25 ft. high, the larger sizes containing a pipe center post.

No mention had been made in the paper of explosion hatches, which were sometimes installed on the tank roof, to relieve the pressure and prevent the tank from being damaged in case of

[1] Fuel Oil Dept., Standard Oil Co. of N. J., New York, N. Y.

[2] Mgr., The Oil Conservation Engine Co., Cleveland, Ohio. Jun. A.S.M.E.

[3] Vice-President and Chief Engineer, Shepard Elec. Crane and Hoist Co., Montour Falls, N. Y. Mem. A.S.M.E.

[4] Industrial Engineer, Day & Zimmerman, Inc., 1600 Walnut St., Philadelphia, Pa. Assoc. A.S.M.E.

lightning fires. A difference of opinion existed as to the real merit of the explosion hatch, but if properly designed and mounted to form a seal when closed, it was considered a good safety feature. Large tanks were usually equipped with eight of these lift doors symmetrically arranged in a circle around the roof.

If tanks could be built to remain tight, notwithstanding the weaving and distortion they experienced from expansion, contraction, and hydraulic strains, the number of losses from electrical storms would undoubtedly be reduced. Some engineers were of the opinion that tank fires were usually due to combustible gases escaping at the joints and becoming ignited when static electricity in the storm clouds and induced charges in the tanks were neutralized, usually with spark accompaniment. The flashback into the tank ignited its contents.

Other tank and pipe details, which the authors of the paper no doubt had omitted on account of space limitations, included suitable gaging facilities to indicate the quantity of oil on hand, twin strainers on suction lines to the pumps, to permit of one screen being cleaned while the other was in use, relief valve on circulating systems to insure constant pressure and continuity of flow, preheating of the oil between pump and burners, suitable pumps for different classes of service, etc.

The local restrictions applying in the district and the regulations of the National Board of Fire Underwriters were of prime importance and should not be overlooked in determining tank locations, arrangement of pipe systems, sizes of equipment, etc.

M. M. Osborne,[1] in the course of the oral discussion which followed, said that several points brought out in the paper were modified in installations which had to be made in crowded cities like New York and Boston. There was more and more use of heavy fuel oil in large downtown buildings and industrial plants situated in the hearts of large cities where there were no rail or water transportation facilities. Under such circumstances, size of tankage was limited not so much by the consumption as by the space available for installing the tankage. Delivery was usually by truck because this was the only available method. Many installations where large quantities of oil were used yearly were absolutely dependent on the delivery of three or four truck loads every day in order to keep going. This was not impracticable in city districts because the streets were open and it was possible to make deliveries at night if not during the day.

In city installations it was sometimes impossible to use heaters in the tanks. New York City did not favor steam coils in tanks in city installations, and it was necessary to put in quite an expensive hot-water circulating system if the tank was situated beneath the floor so that it was surrounded by cold water. Ordinarily the difficulty was overcome by heaters on the suction lines near the tank. The suction heater heated the oil to 100 deg. or so and made it fluid enough to flow to the pumps.

It would be of interest to hear the experience of the oil companies in handling accumulations of heavy sludge at the bottoms of tanks.

Mr. Osborne concluded by calling attention to the fact that there was also encountered in city installations a rule prohibiting openings in the tank below the top, so that connections must be made through the top rather than below as in marine and open-air installations far enough from buildings not to be subject to the Underwriters' rules.

W. W. Oakley[2] spoke of the difficulties in maintaining uniform conditions in combustion when a tank car of oil, containing sludge, was emptied into the storage tank, the sludge distributing itself throughout the oil so that two tanks were necessary. He asked what was the best method of eliminating the water in the bottom of a car, there being as much as three to six inches in some cars. He also wanted to know how to dispose of sludge when it could not be dumped into the river.

Albert A. Cary[3] said that the question often arose as to when it was desirable to substitute oil for solid fuel. One of the first things that he advised his clients to do was to investigate the availability and source of supply of the fuels. Other factors were the character of oil that was available, the problems in city installations to comply with the rules for the use and storage of oil, insurance, and the price of oil. At the time of the World's Fair in Chicago, when they had wanted a "White City," they determined to adopt the use of oil. Oil then was very cheap. Cement mills were using it for burning cement. The price went up as the demand for refinery products increased and the residual oils we depended upon so much at that time were not as available. Mexico oil which was $3\frac{1}{4}$ cents a gallon went up to 5 and $5\frac{1}{2}$ cents. Prices went up to a point where it was as economical to use solid fuel in New York plants. Of course with oil the expense of handling coal and ashes and some of the labor cost was cut, so that oil still might have an advantage at a greater cost.

H. L. Eckerson,[1] recounting the experience of the company with which he is connected with an installation made in downtown New York, said that a change from coal to oil was made because in burning coal it was necessary to use three shifts of men every twenty-four hours. The boilers were located so far from the coal bunkers that conveyors could not easily be installed, so recourse was had to oil. The rules covering the installation of oil-burning plants in New York City were very strict, so strict in fact that it was almost impossible to comply with all of them.

The construction of this plant was particularly difficult as the tank had to be brought in in pieces and welded on the job. It happened to be enclosed in brick walls, which made it difficult to get at, and also it had a division of steel so that one side could be shut off while being cleaned.

The oil was delivered in insulated wagons at a temperature of 100 deg. There was a slight temperature loss in transit and a steam coil was used in order to pump the oil to the boilers. The capacity of the tank was only about 7500 gal. due to the crowded condition of the boiler and engine rooms. Formerly three boilers had been used, but with oil two proved sufficient. The installation paid for itself in about one year, but such savings depended on circumstances. The installation of a conveyor would have made the use of coal cheaper.

H. H. Fleming, who presented the paper, answered many questions put to him during the discussion. In answer to a question by Chairman Shepard, he said that he believed the most economical proportions for small tanks resulted when the diameter was equal to the height. When the diameter got to be 35 or 40 ft. at most, the weight on the bottom ring was so great as to require expensive construction. The largest tanks in common use until recently had been 115 ft. in diameter by 35 ft. high. Within the last few years the size had increased to 120 ft. in diameter by 40 ft. high.

As to the cost of heating and pumping, which Mr. Shepard had asked about, Mr. Fleming said that these varied so much that it was hard to be specific. The pumping depended entirely on the layout of the plant and was not a great expense. Heating was done by exhaust steam from the pumps. The cost would depend on the character of the oil, on whether the entire tank was heated or merely the suction, and whether the tanks were insulated or not. The specific heat of oil was about 0.5 and a heavy grade of fuel must be heated from atmospheric temperature up to about 100 deg. fahr. Certain types of burners would require an additional heating beyond this temperature.

Mr. Fleming said that the economic relation between the amount of storage and the maximum rate of consumption depended entirely on location with regard to source of the delivery and the manner in which the delivery was made. Very roughly, a plant should have about a month's supply and should be able to store such an amount. A tank car contained from 6000 to 12,600 gal., but it was not always possible to get a small-size tank car. If deliveries could be taken in tank cars, traffic conditions should be studied to know how long it would take to get from refinery to plant. With barge deliveries around New York, a 24-hour notice was usually sufficient. Barges handled from small quantities up to about 15,000 bbl. or more. As steel tankage was not very expensive compared with shutting down a plant, it paid to be liberal in planning the amount of storage.

Mr. Fleming took occasion to emphasize the point made by Mr. Moore that a wooden roof was not economical. Very few had been

[1] Asst. to Genl. Mgr., Petroleum Heat & Power Co., 100 Boylston St., Boston, Mass. Mem. A.S.M.E.

[2] In charge of Fuel and Refractory Materials, Corning Glass Works, Corning, N. Y. Assoc-Mem. A.S.M.E.

[3] Cons. Engr., 95 Liberty St., New York, N. Y. Mem. A.S.M.E.

[1] William A. White & Sons, New York, N. Y.

used in industrial plants or in refineries, although until recently many had been used in the producing fields. From now on the wooden roof would be a thing of the past. The rate of insurance was double that for steel roofs. Very few steel-roof tanks were struck by lightning, especially when connected to well-grounded pipes.

With reference to the globe roof mentioned by Mr. Moore, Mr. Fleming said that tanks with such roofs were being built for gasoline in capacities up to 80,000 bbl. The construction was more expensive but the roof was tighter, which prevented gasoline leakage. In Europe tanks of this type had been built up to 60 ft. in diameter. The roof was built first, on the foundation, and then the sides and finally the bottom ring, using no scaffolding.

Mr. Fleming said that in his opinion the explosion hatches mentioned by Mr. Moore were not worth putting into a tank roof. Any hatches to be of sufficient value to take care of an explosion would have to be nearly the area of the whole roof. For storage tanks they were not worth while. While they might save the roof in some cases, they increased the fire hazard as it was almost impossible to make them tight against the leakage of vapor, and at the same time have them work freely enough to be of use in case of an explosion. The commoner practice was to make the seam between the roof and the angle at the eaves the lightest part of the tank, so that in case of explosion it would rupture at that place. Close spacing of light rivets was preferable to wide spacing as the latter tended to leakage. It often happened that tanks containing light oils in process would explode and lift up the roof, and that the roof in falling back would snuff out the fire.

With reference to the point brought out by Mr. Osborne that in cities there should be no connections except through the top, Mr. Fleming said that he considered the rule to be wrong. When a tank was on fire the first thing to be done was to remove the oil, and this could not be done if the connections were at the top where they had been destroyed by the explosion or burned off. With bottom connections the oil could be drawn off into another tank which most installations provided, leaving only a small amount to burn. The question was a much-argued one.

As to taking care of an accumulation of sludge, Mr. Fleming said the best method was not to let it accumulate. While not always possible, the type of burners used in industrial plants would handle a little sludge with the oil. This was done by having a swinging suction pipe which occasionally could be lowered below the sludge line, taking a little of it along with the oil. With some burners this could not be done, in which case the sludge accumulation must be cleaned out.

Heating was of little avail in the case of sludge, he said. Where oil was heated near to the boiling point of water, special care had to be taken not to let the water accumulate. In case of fire, water in the bottom of a tank was dangerous due to the fact that it would boil if the fire got close enough and would cause foaming. A foot and a half or two feet of sludge would foam over a 30-ft. tank under such conditions.

Answering a question by Mr. Shepard about the safe limit of heating oil, Mr. Fleming explained that the flash point was that temperature at which a slight flash appeared when a small flame was held over the surface of the heated oil. This merely showed that some vapor was given off. The general rule in storage plants was to keep the oil below the flash point so that no vapors were given off. In marine installations where mechanical burners were used, the oil was heated beyond the flash point, but only on the discharge side of the pump. It was seldom necessary to get close to the flash point in heating oils for pumping. Mr. Fleming could not understand how there could be three to six inches of water in the bottom of a tank car, as reported by Mr. Oakley, unless it accumulated during steaming out a car. As to sludge disposal at the refinery, he said, some of it was separated and re-run, while some was burned. Considerable work was being done on treating methods to separate sludge. With a fraction of one per cent of sludge in oil, which was a large proportion, it would be some time before the accumulation would be serious.

Application of Engineering Principles to Economic Problems in The Textile Industry

AT THE Annual Meeting of the A.S.M.E. last December, the Textile Division held a session at which Eugene Szepesi[1] presented a paper on The Engineer's Field in Industrial Economics. This paper appeared in the Mid-November, 1924, issue of MECHANICAL ENGINEERING, pp. 841–848. A discussion of the paper follows.

Alonzo Flack[2] submitted a written discussion in which he said that our educational institutions were starting the training of many engineers. Surely the aim must be to improve our industrial developments as to form, methods, and economics. In working toward the betterment of form and methods, the engineer must give consideration to the economics of the situation.

If capital investments were to have a reasonable rate of return, the relation between revenues, expenses and profits should be forecast, and to do this properly required intensive research by one whose time was unoccupied with other immediate considerations. Was not the establishing of the facts upon which this preplanning was based a function of a trained engineer?

Over twenty-five years ago, wrote Mr. Flack, L. R. Brockway had worked out a plan of predetermining the financing of the Elmira State Reformatory, and it had worked. One of our largest industrial companies with the help of trained engineers now controlled the financial activities of over eighteen plants through carefully prepared forecasts which worked out in practice with surprising exactness.

The engineer was an investigator and a creator. He searched for the truth, analyzed the facts, and planned improved or corrective measures. The executive put these plans into operation with help and guidance from the engineer. The executive was crowded with the immediate things to be done and in his hurry left as commonplace much that was fundamental to his own success. The engineer searchingly studied fundamentals for the purpose of devising corrective measures.

Wastes were often considered unpreventable because it had been established by past performance that a certain percentage of waste was about the best that could be hoped for. In many such cases the engineer would establish

1 Many causes for the wastes
2 That dependent sequences constituted an important cause of greater waste
3 That a standard maximum percentage of waste could be arrived at way below what was thought possible
4 That the causes for waste could be taken up one at a time and thus be eliminated.

Standards were a great aid in measuring any performance. It was vital, however, that the standards should be attainable. The engineer after his research was in a position to work out and recommend standards. Standards could be worked out not only for details but for the overall accomplishment in industrial and business institutions.

In planning, one should not confuse predetermining and recording. It seemed unnecessary to call attention to this point, but there appeared to be considerable misunderstanding about it. A correct knowledge of the principles of planning was far more important than the methods employed. Some of the best production-control installations did not require charts or production-control boards. However, if the principle was kept in mind, ways and means for perfecting the method would come to light. With proper pro-

[1] Engr. and Economist, Szepesi Industrial Organization, Boston, Mass. Mem. A.S.M.E.

[2] Vice-President, The Emerson Engineers, 30 Church St., New York, N. Y. Assoc-Mem. A.S.M.E.

duction control, a great aid to forecasting financial control was at hand. A well-known manufacturer was quoted as follows in speaking of the results of planning, scheduling, and dispatching during the trying war period of 1917-1918.

> Between November, 1917, and April, 1918, we had war; we had coal famines; we had embargoes; we had snow blockades; we had labor troubles; we had a combination of everything that would preclude a man from keeping his promises. . . . In those six months we wrote our customers and made absolute promises of shipment in 7666 cases, and out of the total there were just six promises broken.

This was the result that engineers could help the management in bringing about, both as to financial and production control.

It was not only necessary to set up standards and plan to attain them but those who must do the work must be taught and appealed to emotionally and fairly. The engineer by training and experience could be of great assistance to the operating executive in working out the plans and methods for interesting the great mass of workers, who must be constantly helped and encouraged.

McRea Parker,[1] who opened the oral discussion of the paper, said that the author, Mr. Szepesi, deserved a great deal of credit for what he had done, for one of the hardest things to do was to go into a textile mill and attempt to introduce the procedure outlined in the paper.

In the matter of mending imperfections, what the author said was true. Profit was either made or lost there to a large extent, and this had caused many mill operators to look further back in the production with a view to cutting down the costs in this particular department. The author had emphasized how this cost of mending might be determined. It was not within the scope of the paper to bring out that many of the faults could be eliminated in some previous process, but Mr. Parker thought that the mechanical engineer going into the factory would see the need of this kind of an analysis. The men in this particular industry who had been in it for years and had made a success of it in the past three or four decades did not realize that within the next twenty or thirty years they would have to run mills differently.

While it was not quite clear in his discussion, the author had set many of his costs on time unit studies. This was another very difficult thing to persuade management in the textile industry to adopt because of the large varieties involved; however, if more mechanical engineers would invade the textile industry, they would bring about results in kind.

H. M. Burke[2] said there was one point that might be brought up to show where the mechanical engineer might become an economist in the textile field, namely, in helping to eliminate the idle time of equipment. The author had shown where the production could be controlled, but one of the greatest drawbacks was the scientific management of the maintenance of the equipment. He had not shown whether he had taken into account idle equipment, and Mr. Burke thought this problem could be attacked profitably.

Mr. Szepesi, replying to Mr. Burke, said that it was necessary to consider idle equipment for several reasons, as well as burden or overhead cost. The cost of standard operations of a mill was determined when it was operating at 100 per cent efficiency, or at 80 per cent, which was nearer the true condition.

There were, he said, certain instances where machinery was idle on account of carelessness or lack of inspection. This was brought to the attention of the management in the form of machine-hours, as everything was considered on an hourly basis. Assuming, for example, a spinning department with 30,000 spindle-hours a week and orders enough to require all of them, the standards set would be for 30,000 spindle-hours a week and actual operations would be compared with that. If it were found that the department operated only 27,000 spindle-hours on account of breakdowns, it would be determined at the same time that it operated only 25,000 spindle-hours on account of inefficiency of operation. This was one of the most important points to be considered, because the burden cost was going to be determined by the number of spindles and the number of looms.

Mr. Burke, resuming, said he thought from the viewpoint of seventeen years' experience in several kinds of textile plants that not only the management had been lax along the lines of elimination of idle equipment, but also the mechanical engineer himself; and he believed that the mechanical engineer could go into the study of equipment so that less of it would be needed for the same amount of production, and that he could aid the economies immensely by making a more scientific study of the equipment and some of its features, such as lubrication.

Mr. Szepesi agreed with this and said that there was another phase of it illustrated in the loom department. A standard allowance was made for breakdowns of a given type of loom in which individual looms were not considered but a group of looms under the charge of a loom fixer. Then, if the production was less than standard, the equipment was investigated first to find the reason for the difference. It was often found that the looms had not been carefully repaired. In one case more than seven picker sticks had been found broken in one loom; the fixer had put in new sticks but had not eliminated the cause of the trouble.

Time was very important and should be considered in the cost of manufacturing because the burden lost in idle equipment when there was still work in progress was a total loss to the management, and no business should have idle equipment while there were orders upon which to work.

E. J. Flather[1] said that the troubles of many in the textile industry were not due to oil or picker sticks but to the amount of idleness due to lack of orders. That industry had been asleep for a hundred years. It was the first one to which power had been applied. It was the oldest industry, and it was the one which today gave more for the dollar than any other, he believed, in the United States. Engineers had not yet found how to overcome the disadvantage of a greater labor cost of 25 per cent, to state it moderately, in Massachusetts. Others with lower costs were making exactly the same styles elsewhere and almost as perfectly, so that they could supply 90 per cent of the goods required. If engineers would tell how presidents, treasurers, agents, superintendents, overseers and employees in Massachusetts could be made about ten per cent more efficient, the concerns in that state might manage to keep the wolf from the door a little longer.

Charles T. Plunkett[2] said that the real coöperation desired was that which Mr. Flather had spoken of. The textile manufacturers wished the assistance of the engineer in so planning their machinery, so fixing their schedules, and so shaping their markets that they might have continual operation in their mills under the adverse conditions which prevailed in Massachusetts in short hours, the employment of women after six o'clock, and other matters. He thought that Mr. Flather was quite within bounds when he said that ten per cent increased efficiency in Massachusetts mills would enable them to compete with those of the South.

Mr. Plunkett declared that he was not one to be stampeded by Southern competition. He thought there were decided advantages there in certain fabrics. There were disadvantages too, some of which mills in the North did not have, and if the difference in the cost of labor, which was practically 50 per cent higher in Massachusetts than in the South, were reduced to 10 or 15 per cent, it would hardly worry most of those who had good plants in the North. The problem for engineers, as Mr. Flather suggested, was to enable the mills in the North to improve possibly ten per cent in their efforts to overcome these seemingly great disadvantages.

Mr. Flather had spoken of the lack of improvement in textile usages, and this was literally true. There had not been a great many in the last hundred years. However, he was the surviving member of a partnership which had continued in the textile business for ninety-five years and so necessarily had had a rather long experience, during which they have tried not to become so old-fogyish as not to keep up with modern inventions and improvements, regardless of their source. They had looked largely to the textile man to make these improvements. He believed in calling in the mechanical engineer, for it was quite possible to make their greatest advances under suggestions of men who were not supposed to understand the industry in particular. They did look to the mechanical engineer for the solution of many of their technical problems, and

[1] M. and E.E., Cleveland Worsted Mills Co., 6114 Broadway, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

[2] Plant Engineer, Mt. Hope Finishing Co., North Dighton, Mass. Mem. A.S.M.E.

[1] Mechanical Engineer, Room 1119, 40 Court St., Boston, Mass. Mem. A.S.M.E.

[2] President, Berkshire Cotton Co., Adams, Mass. Mem. A.S.M.E.

though they might be unfamiliar with the entire question of technical manufacture, here and there they would discover places where their suggestions might be more valuable than any others to the textile manufacturer.

J. W. Cox, Jr.,[1] said that systems of cost accounting had had somewhat of an unfortunate experience in the textile industry. The reason was that most of the men who had had to do with them had been accountants and their systems were often based on opinions rather than facts. They had had elaborate systems of figures, records, papers and other things which cost a lot of money but from which nothing had been quickly available.

The author, Mr. Szepesi, had based his system on facts which had been obtained by investigations. These facts had been given in such a comprehensive manner that immediate use could be made of them. In one case where they had been applied, the number of clerks had been reduced 75 per cent and the work performed better. Instead of employing themselves in correcting a lot of troubles, the operators had had time to plan how to avoid the troubles. If a business was based on facts immediately applied, it would produce better goods at lower costs and have a larger volume of business.

F. S. Blanchard[2] said that in one of the mills of his company a research organization analyzed a job to find out how an operative spent her time and found that about thirty per cent was spent in weaving, about forty-five in bobbin filling, and the remainder in sitting down. The other jobs were then analyzed and it was found in one that fifty-five per cent of the time was idle. As a preliminary to bringing about economies it was necessary that the machinery be put in good condition and then carefully used. Under the new basis of operation, the wages fitted the job; the weaver was weaving only and some one else was filling bobbins, and skilled labor could be paid more as a result. New England manufacturers were too conservative. They thought these things could not be done. It was perfectly possible to cut labor costs by other means than by reducing wages.

Mr. Szepesi, in closing, said that, referring to Mr. Flather's remarks on increasing production 10 per cent by greater efficiency, the previous last year he had developed a table for the manufacturer which would reduce the overhead by 5½ per cent. The engineer would have means of increasing individual production through the coöperation of the worker if the worker would coöperate, but in some cities it had been found that the number of looms to be cared for by a loom fixer was set under labor-union conditions and conceptions of efficiency. He believed that if the manufacturers of Massachusetts were to go to their operatives and try to imbue them with a sense of their responsibility by presenting the facts, they would be rewarded. The worker should understand that if his personal efficiency was decreased, the increase in overhead was going to wipe out the profit of the mill. This was a problem on which he was spending much time. He did not believe one man could solve it, but he was convinced that if Massachusetts manufacturers were to spend more time with their workers, presenting them the facts and getting their good-will, it would greatly increase efficiency. Every manufacturer must admit that there was much to be done in a psychological way. All had problems in reducing the cost and increasing the efficiency, but mechanical devices entirely would not suffice.

Radical things could not be expected in the textile industry, but it was possible to eliminate small wastes of time and effort, which would ultimately bring down the cost of manufacturing, and to do this it was necessary to have standards of measurement to show what was the best way.

That, said Mr. Szepesi, was probably all that he could prophesy for the textile industry. He thought all would agree that it was the only solution. It was not legislative. There would always be men in office who were not interested in the manufacturer but in the people's votes. It was always possible to "knock" a manufacturer and show the profit he had made. The manufacturers in the textile industry, especially in the cotton industry, should make an effort to study intelligently the psychological factors of the worker. Not the welfare, not the housing conditions, not showing how wonderfully they had improved the tenements, but what efforts they had made to get the coöperation of the worker. The men in the office did not understand the temper of the man at the machine, but to strangers they told many things they would not tell to the manufacturers.

Oil Burning Under Steam Boilers

Discussion Dealing with Atomizers, Draft, Oil Characteristics, Refractories, Heating of Oil, Combustion Data, Hazards of Industrial Oil Burning, Etc.

A SESSION on Oil Burning was held at the recent Annual Meeting of the A.S.M.E. in New York under the auspices of its Fuels and Power Divisions, E. H. Peabody[3] presiding. Three papers presented, at this session, one by H. G. Donald[4] on Fuel-Oil Burning in the United States Navy,[5] which discussed the type of atomizer, combustion data, draft, boilers, furnaces and refractories, characteristics of oil, atomization and pumping and heating of the oil; another, by Nathan E. Lewis,[6] on Oil Burning in Industrial-Plant and Central-Station Service,[7] which took up steam-atomizing vs. mechanical-atomizing oil burners, oil heaters, furnace volume, introduction of air for combustion, and firebrick problems due to high furnace temperatures, and a third one, on Hazards of Industrial Oil Burning,[8] by H. E. Newell,[9] in which the author discussed such questions as the flash point, heating of oil in storage tanks, storage methods, fire-hazard features of design, installation and operation, and causes of oil fires. The discussion of these papers follows.

[1] Cons. Textile Engr., 320 Broadway, New York, N. Y. Mem. A.S.M.E.
[2] Pacific Mills, Lawrence, Mass.
[3] President, Peabody Engineering Corp., New York. N. Y. Mem. A.S.M.E.
[4] Lt-Com. U. S. N., Fuel Testing Plant, U. S. Navy Yard, Philadelphia, Pa.
[5] Published in MECHANICAL ENGINEERING, Mid-November issue, 1924, p. 769.
[6] Mechanical Engineer Babcock & Wilcox Co., New York, N. Y. Mem. A.S.M.E.
[7] Published in MECHANICAL ENGINEERING. Mid-November issue, 1924, p. 849.
[8] Published in MECHANICAL ENGINEERING, Mid-November issue, 1924, p. 765.
[9] Engineer, National Board of Fire Underwriters, New York, N. Y.

FUEL-OIL BURNING IN THE U. S. NAVY

Joseph J. Nelis,[1] discussing Commander Donald's paper, said that the problem on the ship, both in the Navy and in the merchant marine, was entirely different from that on land in that there was a fixed maximum load. Boilers could be designed for that load. If the merchant marine, ships were operated at practically full speed all of the time, and as there were no peaks, the boilers could be operated at the loads for which they were designed.

As he understood it, in the Navy full speed was only an occasional condition, and although the fuel-oil testing plant had operated this White-Forster boiler at around 22 lb. per sq. ft. of heating surface, such a condition aboard a naval vessel was seldom required. When it was required, however, it was badly needed.

At very high ratings it had been determined that the furnace efficiency was limited by the amount of fuel that could be burned without burning out the brickwork. On ships they were trying to burn a great deal of oil for a very small surface, and a steel-cased furnace with no chance for air to get at the brickwork was promptly burned out.

As regarded high capacity, he believed that the highest capacity

[1] Secy., Forster Marine Boiler Corp., 111 Broadway, New York, N. Y. Mem. A.S.M.E.

ever obtained either with oil or any other fuel had been obtained with the locomotive boiler.

Replying to Mr. Nelis, Commander Donald said that the question which he had brought up involved the reduction of furnace temperature by the installation of superheaters of what Commander Donald called the radiant-heat type in order to reduce the furnace temperature. At the Fuel-Oil Testing Plant there was now being conducted a test of Mr. Nelis' superheater installed in the back of a Yarrow boiler. While these tests were not yet completed, he would say that it was necessary to have the temperature in the furnace in the neighborhood of 2700 to 2800 deg. fahr. in order to support a high rate of combustion. Otherwise heavy vibrations or pulsations occurred. The chief difficulty in burning oil, especially in marine furnaces where the amount was so great per cubic foot of furnace volume, was in obtaining a sufficiently long flame to keep the excess air down to a reasonable figure, and still not have the flame chilled by the water-heating and superheating surfaces to the extent of forming smoke. Smoke was taboo.

The furnace temperatures prevailing now were as high as 3000 deg. fahr. with 11 lb. of oil per cu. ft. A refractory only $4\frac{1}{2}$ in. thick was used, and in express-type boilers in destroyers in the past there had been no insulation except a 2-in. air space and a light sheet-metal covering on the outside.

Walter M. McFarland[1] submitted a written discussion of Commander Donald's paper in which he pointed out that the author's modesty had probably prevented him from mentioning the work carried on at the Fuel-Oil Testing Plant, of which he was the head, and the value of this work to the country at large as a peace-time contribution of the Navy. The experimental and development work done at this plant had been very valuable and the establishment of standards in firebrick, insulating material, etc. had been of real service on land as well as sea.

An example of the experimental work carried on at the Fuel-Oil Testing Plant was the blower burner (Fig. 1) to which Commander Donald had referred. This burner was the invention of Com. H. H. Norton, U. S. N., Commander Donald's immediate predecessor as head of the Fuel-Oil Testing plant. The success of this burner had led to its adoption as standard equipment for all U. S. naval vessels, and it was interesting to note that all oil-burning equipment on our battleships and the new 18,000-s.hp. airplane carrier was being converted to the use of this type of equipment exclusively. In addition some of these burners were being installed on each boiler of every destroyer and scout crusier.

One of the great advantages of this type of burner was the remarkable saving in fuel at low speeds and particularly at port load. In a recent issue of the *Journal* of the American Society of Naval Engineers, Com. A. M. Charlton, U. S. N., described extensive tests run with this burner on the U. S. S. *Billingsley*. These tests showed a saving of 25 per cent in fuel consumption at port load. Later tests showed a saving of 20 per cent at 12 knots speed.

These savings were due not only to the efficiency of the burner but also to cutting out the large blowers used to put the fireroom under pressure.

The steam consumption of the small turbo-blowers used with the burner averaged about $1\frac{1}{2}$ to 2 per cent of the steam generated, but this steam was returned to the system, so that the actual heat loss was a small fraction of one per cent.

The Fuel Conservation Committee of the Shipping Board was among the first to appreciate the possibilities of this blower burner and as a result of its recommendations, several installations had been made in Shipping-Board vessels with good results.

Several installations of the blower burner under the name of the "Mayflower Burner" had been made in stationary plants where the capacity of the boilers was limited on account of draft. The results had been most satisfactory. The elimination of ducts and large forced-draft fans with a corresponding reduction in installation charges, as well as the high efficiencies obtained with this burner, were important considerations in favor of this type of equipment for some stationary installations.

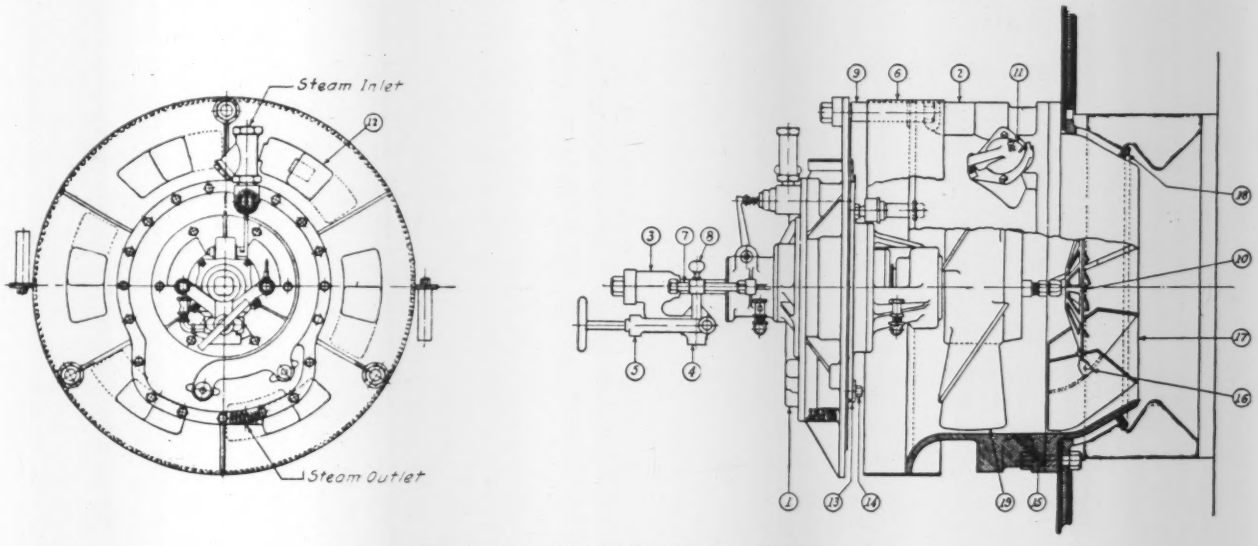

FIG. 1 CROSS SECTION OF BURNER BLOWER

(1) Turbine; (2) air-inlet casting; (3) atomizer; (4) oil-burner coupling; (5) coupling yoke; (6) air shutter; (7) coupling studs; (8) thumb screw; (9) turbine-supporting studs; (10) impeller plate; (11) peep holes; (12) radial shutter; (13) shutter guides; (14) tap bolts for shutter guides; (15) studs for air inlet casting; (16) impeller plate locking clips; (17) bladed cone; (18) burner brick locking grid; (19) fan.

Since boilers were operated at high ratings in stationary practice today, it was of considerable interest to note that the test on the White-Forster boiler at the rate of 1.5 lb. of oil per sq. ft. of boiler heating surface referred to by Commander Donald was equivalent to about 646 per cent of boiler rating on the A.S.M.E. standard.

C. J. Jefferson[1] wrote that about two and a half years ago a Fuel Conservation Committee appointed by the U. S. Shipping Board, consisting of representatives of the various interested engineering societies, including the A.S.M.E., had started to attack the problem of reducing the fuel bill of the Fleet Corporation vessels.

In studying this problem it became evident that the point where the greatest benefits could be obtained was in interesting the operating personnel themselves in the problem and giving them the training which would enable them to attack it in a logical manner.

To this end arrangements were made whereby the Navy school was opened to the engineer officers of the American merchant marine, and during the past two years some six hundred officers had attended the Philadelphia Navy Yard school; and last year, at the urgent request of operators on the Pacific Coast, a similar school had been established at Mare Island Navy Yard, San Francisco, where up to date about one hundred officers had been trained.

Mr. Jefferson wished to bring out this point, for while all knew of the mobilization plans which the Ordnance Department of the

[1] Manager, Marine Dept., Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. Mem. A.S.M.E.

[1] Head of Fuel Conservation Section, U.S.S.B., Philadelphia, Pa. Mem. A.S.M.E.

Army had been developing among business men and engineers, he did not believe that sufficient thought had been given to that most important of all factors in time of national emergency, namely, the plan of mobilization of operating forces for our vessels, and he felt that this school would serve in a way as a nucleus around which could be constructed a strong program for carrying out such a plan.

R. Warriner[1] wrote that what were now called "individual blower burners" were no doubt useful and economical where one or two were fitted on each boiler primarily for use at very low speeds or at anchor, but it was doubtful if a complete installation would prove satisfactory under all conditions. For an express-type boiler of, say, 10,000 sq. ft. of heating surface, there would be seven or nine such individual blower burners which meant seven or nine small turbines with their attendant steam, exhaust, and drain piping, all of which had to be taken across the front of the boilers. In addition to this, there would be of course the regular blowers, though these might possibly be of smaller size than at present.

In regard to the air pressure used in the fireroom, when using express-type boilers and high rates of evaporation it was not considered very desirable to have a low air pressure. It might be possible to burn the oil satisfactorily with lower pressures than were now in use, but the higher air pressure, namely, 8 or 10 in., formed a good factor of safety, and greatly reduced the risk in the fireroom of damage to the personnel from burst boiler tubes.

On the score of efficiency it was not considered that 100 per cent blower burners plus the regular blowers running at reduced capacity would be as good for moderate or high speeds as the installation now in use.

In answer to several questions by E. Y. Sayer,[2] Commander Donald added the following explanations to his paper.

The draft of 0.3 in. which he mentioned was measured at the base of the smoke pipe.

The blower type of burner was turbine-driven, although motor drive would be more desirable. The Ray burner had a motor drive and a different method of atomization, but a similar method of supply of air to the burner. The turbine had nothing to do with the atomization of the oil and was used only to create a draft.

In answer to a question by H. T. Dyer[3] as to why the percentage of CO_2 should be less with the larger types of burners, Commander Donald said that while he had had very little experience in conducting tests with varying CO_2, he thought that the explanation was that the length of the flame tended to be necessarily longer in the larger burner and that it was therefore necessary to increase the excess air in order to prevent smoke.

The oil companies should be required to furnish necessary information to the users of their oil so that it could be burned properly. They should give its viscosity curve as it was not easy for every individual industrial plant to work out one for its own use.

In the Navy, he said, the tankers and capital ships were provided with viscosimeters and there was no reason why the information should not be given to all the ships, although it never had been.

In buying oil, he said, such as Bunker C, it was probable that one carload might be Bunker C and the next Bunker A. He had found that he could buy Bunker A at the same price as Bunker C and that although the B.t.u. value of Bunker A was lower, Bunker C required more preliminary heating, so that all in all, Bunker A might be more economical than Bunker C.

OIL BURNING IN INDUSTRIAL PLANTS AND CENTRAL STATIONS

E. B. Powell,[4] in discussing Mr. Lewis's paper, wrote that, under the conditions of high and varying-load demand frequently confronting the central-station boiler plant, the mechanically atomized-oil-burning furnace of solid-wall construction had shown very short life, rarely more than five or six months for the average first-quality firebrick. The cause of this rapid failure was doubtless to be found in the intense concentration of heat which was characteristic of this mode of combustion, coupled with the absence of any form of protective slag coating on the face of the brick. The great preponderance of the failure is in the nature of "spalling," the more or less gradual cracking, chipping, or crumbling disintegration of the brick. Just what qualities the brick should possess to resist the action of spalling was so far impossible to state in definite terms. The bricks which had given best service might be grouped approximately in three general classes:

1 Bricks of high fusion point and inherently high resistance to spalling.

2 Bricks of high fusing point which, while subject to fracture under conditions of repeated temperature change, were of such highly refractory material that there was no tendency to softening even at the maximum temperature of the furnace, so that parts broken off did not become dislodged but remained to serve as a protecting screen for the main body of the brick.

3 Bricks which had comparatively low inherent resistance to spalling and combined with a comparatively low fusing point great tenacity in the semi-plastic state. Bricks of this class, if employed within the limits of their safe temperature range, glazed over to form almost monolithic walls and gave a longer life than many bricks of much more highly refractory material. However, such bricks softened rapidly under temperatures but slightly higher than those for which they were best adapted, and should be used with caution where subjected to heavy loading.

As the author had pointed out, boiler settings for oil-burning plants were now being constructed with ventilated furnace walls as a means toward reducing the rate of brickwork deterioration. Up to the present time these settings had been designed for operation with the ventilating ducts at a pressure slightly below atmospheric, that was, under suction from furnace or fan. His company was now placing in operation a number of settings in which the ventilating ducts would be supplied with air under pressure of the forced-draft fans. This arrangement should provide even more effective cooling of furnace brickwork and should have the further advantage of avoiding the necessity for high pressure differences on the walls of the setting beyond the ventilated area.

In connection with studies of boiler setting, materials, and design, his company had made some observations to determine temperatures to which furnace walls were subjected under normal operating conditions when burning oil with the mechanically atomizing burner, the furnaces in which the observations had been made had approximately equal rectangular horizontal sections and areas of heating surface exposed to direct radiant heat, but were of different heights from the floor to heating surface. The smaller furnace had a total refractory surface (including floor) per square foot of exposed heating surface of approximately 3.4 sq. ft. and a volume of approximately 7 cu. ft. per square foot of exposed heating surface—the exposed heating surface being figured as a plane surface parallel to the axis of the tubes limited by the four walls of the furnace. The larger furnace had corresponding refractory surface and volume of approximately 5.4 sq. ft. and 10.7 cu. ft., respectively. It had been found that when burning as high as 40 to 45 lb. of oil per sq. ft. of exposed heating surface in each furnace using the same atomizing pressure of about 175 lb. and the same rate of air supply, about 20 per cent excess over theoretical requirements, maximum flame temperatures of about 2850 to 2900 deg. fahr. and 2900 to 2950 deg. fahr. were obtained in the respective furnaces, the corresponding average flame temperatures appearing to be about 150 deg. lower in each case with the corresponding wall temperatures reaching about 2600 and 2700 deg., respectively, except that where the hottest flames played against the wall the immediate area might attain temperatures 100 deg. higher than these figures.

A 35-per cent reduction in rate of combustion, if pressure of atomization and ratio of air and fuel remained the same, would in the case of the larger furnace reduce the flame temperature by scarcely 50 deg. and wall temperatures but slightly more, while in the smaller furnace, although the flame temperatures appeared to be lowered only slightly more than 50 deg., the wall temperatures were reduced by 100 to 150 deg. Apparently the higher ratio of absorbing surface to radiating surface in the smaller furnace caused the latter to give up heat more readily than in the case of the larger furnace, so that for a given variation in rate of combustion the wall-temperature fluctuations were greater in the smaller furnace. Just how

[1] Ch. Engr., Bethlehem Shipbldg. Corp., Fore River Plant, Quincy, Mass.

[2] Pres., E. Y. Sayer Engrg. Corp., Knickerbocker Bldg., Broadway and 42nd St., New York, N. Y. Mem. A.S.M.E.

[3] Peabody Engrg. Corp., New York, N. Y.

[4] Cons. Engr., Stone & Webster, Inc., 147 Milk St., Boston, Mass. Mem. A.S.M.E.

closely these temperature conditions would obtain with the same ratios of absorbing and reflecting surface applied to furnaces of greater dimensions, or of different shape, it would be difficult to state without the aid of more comprehensive data than were now available.

Kingsley L. Martin[1] wrote that the custom had been in the past when comparing the different types of fuel-oil burners, to speak and think of steam atomizing burners as producing flat flames suitable only for use over checkerwork, and of mechanical atomizing burners as producing hollow conical sprays to be used with air-control registers.

This assumption had led gradually to a mental confusion, revealed in practically all discussions which had appeared in the technical press recently as to the relative merits of steam and mechanical atomizing of oil for power-house work, which resulted in conclusions being based, not on the method at atomizing the fuel, but on the use or non-use of air registers.

The development of a steam atomizing burner which produced a conical spray identical in shape with the mechanical spray made it necessary to change entirely the previous basis of discussion and radically modify the conclusions reached. The basis of comparison was no longer the shape of the spray, but was the character of atomization produced by the two methods and the resultant operating conditions and requirements.

It had been repeatedly pointed out that with the flat flame heretofore characteristic of the steam atomizing burner, it was possible in a given furnace to use only the number of burners which would fill the furnace with flame from side wall to side wall. The limit as to the number of burners was then reached, it being impossible to double-deck the burners as the upper tier would not receive sufficient air for proper combustion. When air registers were used the air for each burner was drawn in around the burner and each unit was practically independent of the others as regarded air supply, and the number permissible was limited only by the area of the furnace front. If registers were used, with conical flames, it was evident that the air-supply conditions would be the same and an equal number of registers and burners might be used, whether the spray was mechanically or steam atomized.

It had also been contended that 200 per cent of rating was practically the limit of boiler power that could be secured with steam atomizers over checkerwork, as forced draft could not be used to advantage. This was because the flame extended from side wall to side wall, covered the checkerwork, and was so shaped that the incoming air reached only its lower part. More oil meant a long flame, injurious to both boiler setting and generating surfaces. It also meant that as the flame thickened, the air could not penetrate it properly, and incomplete combustion and heavy smoke were the result. With air registers, however, as already pointed out, each unit was independent and either forced or induced draft increased the air supply equally to each burner. But this was a difference between checkerwork and register method of air admission and was not a matter of atomizers, if both produced a conical spray and were used in air registers.

The development of the steam-atomized conical flame had not only eliminated the detrimental characteristics of the flat steam flame mentioned above, but had secured the advantages of the mechanical flame, and had added several marked improvements over the mechanical cone.

This development had opened up an entirely new and extremely interesting field in oil burning. A steam atomizing burner has been designed and perfected to meet many objections to the use of fuel oil and to eliminate the various factors which produced inefficient results.

First, the oil was supplied to the burners at a much lower pressure. Twenty pounds was ample where mechanical atomizers required 150 to 200 lb. to reach the same capacity. This meant low-pressure pumps, strainers, and heaters. Ordinary piping could be used and dangers from leaks were practically removed.

It was not necessary to preheat the oil for steam atomizers as high as for the mechanical. The safety factor in hotels, office buildings, and department stores secured by using comparatively cool oil under low pressure as against oil frequently heated above the flash point and under high pressure, did not need to be emphasized.

The steam atomizer was constructed with large oil passages—the smallest being $^3/_8$ in. in diameter. This simplified the strainer problem, as dirtier and heavier-gravity oil could be used with no fear of the burner's choking up. A suction strainer alone was sufficient, a discharge strainer being unnecessary. A single heater was sufficient, and this might be by-passed if necessary and the oil burned as it came from the storage tanks.

The steam atomizers would run for days without removal for cleaning, and unquestionably a high uniform efficiency was secured by efficient atomizers under steady furnace combustion conditions extended over a long period without the cutting out of a burner.

One of the most valuable features of the steam-atomizing register system was its wide flexibility coupled with simplicity of control. All mechanical atomizers changed the angle of the oil cone with changes in pressure at the orifice, and the diameter of the orifice being constant, a change in capacity of a given tip could only be produced by a change in orifice pressure. In the steam-atomizer register system, however, the oil-cone angle was fixed by the angle at which the orifices were drilled, and the atomization was completed before the oil issued from the atomizer. By varying the oil and the steam from master valves for either a single furnace or a battery of boilers, the burner capacity might range from a pilot light up to 250 or 300 per cent of rating without change in the flame angle, the position of the atomizer, or the register adjustment. The burner had a range of capacity covering the ordinary working range of a boiler in service through which it was unnecessary to change the steam for atomization, so that automatic regulation of the oil alone might be used and the thousand horsepower or more might be kept "floating on the line."

A most important development in connection with the system was its remarkably low draft requirements. Mounted in the *same* air register, the steam cone required but from one-third to one-tenth the draft for the same capacity that the mechanical atomizer did.

One example was an installation of two 600-hp. boilers which required 1.50 in. of air pressure to develop 250 per cent of rating, using the mechanical system, while the steam-atomizing register system carried the same rating with the same results on the 0.20 in. of natural draft that was available. Four registers and burners were used in each case.

This feature meant that many boilers could be operated at 250 per cent of rating with the stack draft now available, and eliminated the more complicated equipment and expense of upkeep inseparable from the installation, maintenance, and operation of engines, fans, ducts, and special boiler fronts common to the mechanical system. If however, forced or induced draft was available and the natural draft was not sufficient to reach the rating desired, the apparatus might be used equally well with the steam-atomizing register system but with a much lower air pressure at the register than was necessary with the mechanical burner.

An illustrative test showing the wide range of capacity, simplicity of control, efficient operation, and reliability had been made in a New York plant on a 600-hp. B. & W. boiler using water-gas tar. The registers and burners developed an average of 230 per cent of rating for 24 hours with only 9 lb. (gravity) tar pressure, 0.15 in. draft (natural) in furnace, 14 per cent CO_2, and no smoke. The maximum rating during the run was 270 per cent with 0.25 in. draft. The boiler was taken from 75 to 230 per cent of rating in a few minutes by operating the boiler tar-supply valve only. During a month's continuous operation there were no carbon deposits and the burners were not removed from the registers. The flame had all the characteristics of a mechanically atomized flame, filling the furnace completely with a soft, low-velocity flame that would run practically indefinitely without injury to either tubes or furnace linings.

In a recent test a 625-hp. B. & W. boiler had developed 1600 hp., or approximately 250 per cent rating, using four steam-atomizer registers on forced draft. The furnace volume was only 955 cu. ft. so that 1.6 hp. per cu. ft. was developed. The oil pressure was 78 lb. and the air pressure at the register only 0.98 in. with negative draft in the furnace of 0.05 in. No attempt was made to see if higher ratings were obtainable.

[1] Pres., The Engineer Co., 19 Battery Pl., New York, N. Y. Mem. A.S.M.E.

G. H. Peltz[1] gave briefly the results of recent tests on blowing cold air on the outside of refractory brick in order to keep the brick from melting. These showed that starting with stationary air the temperature of the outside of the brick dropped very rapidly with even a very slight motion of the air, a motion so slight that it could not be read with a pitot tube. With a furnace temperature of about 2700 deg. fahr. the outside temperature of an 8-in. brick was about 800 deg. in still air. A slight movement of the air reduced the temperature to about 200 deg. But the surprising point was that as the velocity over the brick was increased, the temperature dropped a certain amount and then stopped. At a velocity of 50 ft. per sec. the outside temperature of the brick dropped to 20 deg. above room temperature, and at a velocity of 150 ft. per sec. it dropped no further. The question, then, was why a wall cooled on the outside lasted longer than one which was not cooled. The answer was that whereas the interior surface was still as hot as it was before and while it might be at a point where it was soft and beginning to run, if the brick further back could be cooled, it would have sufficient body to prevent it from falling to pieces.

C. J. Jefferson said that on the S. S. *Leviathan* there was a current of air under the floor of the furnaces heated to about 350 deg., and it had been noticed that the life of the floor in the *Leviathan* was longer than that in other ships in which the rate of combustion was practically the same. The movement of air past the refractory cooled the brick and lengthened its life.

F. M. Gibson[2] said that there had been misunderstanding about the loss of tubes in oil-burning plants. In many plants with which he was familiar, the cause of the trouble had been that feedwater treatment had not been increased in proportion with increased rating, and the loss of tubes had been blamed on the use of oil.

A more serviceable packing for high-temperature and high-pressure oil burners would be acceptable and a tip less susceptible to scouring would aid very much in the automatic control of combustion.

It was customary for oil refiners to induce engineers to accept a heavier grade of oil with a plea that it had a higher heating value. From his experience, comparing oils of 15 with those of 10.5 deg. B., the increased B.t.u. per dollar of the heavier over the lighter grade ranged from nothing to 7 per cent. One point in favor of oil was the accuracy with which it could be measured. This made the records of daily performance more dependable.

K. L. Martin, in answer to a question by T. B. Stillman, Jr.,[3] said that the steam required for atomization amounted to 1.5 to 2 per cent of the steam generated.

As to the effect on the brickwork, he had never been able to detect any difference between a hollow-cone flame atomized in one way or another. In other words, the oil burned in exactly the same way and with the same characteristics regardless of whether it was atomized by steam or pressure.

W. Trinks[4] thought that the Society was neglecting the industrial furnace. There were at least ten industrial oil furnaces for every oil burner in this country. In open-hearth furnaces he had repeatedly measured temperatures of 3250 deg. fahr. There, of course, silica brick was used, which could not be used where the temperature changed quickly. If there was a slight draft in a furnace the flow of air from the outside cooled the brickwork much more than if there were no air flow. If, on the other hand, there was a pressure in the furnace, there would be not only an outflow of heat but of gases and the brickwork would become soft nearer the outside. The operators of industrial furnaces knew this very well and they tried to get their furnaces glazed over as soon as possible in order to make them practically impervious to the flow of gases.

H. L. H. Smith[5] said that in stationary practice, 300 per cent of rating was considered high and operators did not approach it more often than they could help even though their furnaces were large. It might be that in stationary practice high heat efficiency was more essential than in the Navy, and he would like to be informed on this subject.

[1] Fuel & Furnace Engr., Corning Glass Works, Corning, N. Y. Assoc-Mem. A.S.M.E.

[2] Plant Engr., American Sugar Ref. Co., 47 Granite St., Boston, Mass. Mem. A.S.M.E.

[3] Marine Dept., Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. Assoc-Mem. A.S.M.E.

[4] Prof. M.E., Carnegie Inst. of Tech., Pittsburgh, Pa. Mem. A.S.M.E.

[5] Ch. Engr., N. E. Power Co., Worcester, Mass.

Commander Donald said that the life of the brick at high temperature was undoubtedly affected by the insulation on the outside and the thickness of the refractory. The more insulation put on it, even with the same furnace temperature, the quicker the brick would fail. In the boilers of which he had spoken they had burned as high as 10 and 11 lb. of oil per cu. ft. of furnace volume as against 2 and 3 in stationary practice. They had a 4.5-in. refractory with no insulation on the outside other than a 2-inch dead-air space. The air pressures in the fireroom were as high as 10 in. of water. In stationary practice the furnace pressures were much closer to the outside air pressure and there was a large amount of insulation on the refractory.

E. B. Powell said that the longer life of the Navy firebrick was undoubtedly due to the fact that the average temperature of the brickwork was much lower in Navy installations in spite of the fact that under the high rates of evaporation the temperatures in the boiler might be even higher than those in a stationary furnace.

In experiments which his company had been conducting on the dissipation of heat through firebrick, it had been noticed that a temperature of 250 deg. could be maintained on the outside of ventilated although slightly insulated furnace walls, with a temperature of 2900 deg. on the inside. As soon as the furnace was shut down and the ventilating air was turned off, the superficial temperature of the brick would rise to about 500 deg. fahr.

D. S. Jacobus,[1] said that in the absence of Mr. Lewis he would say something on the points that had been raised. He said that Mr. Powell's organization was testing the effects of different slags on different kinds of brick in addition to the tests that had been cited. This kind of work would be invaluable in the solution of the problem under consideration and the organization should be commended.

The same was true of the work on the air cooling of walls mentioned by Mr. Peltz. The thickness of the walls would have a marked effect upon the results of air cooling. A furnace wall for powdered fuel eroded with certain grades of coal and sometimes the air-cool wall was very thin before it failed, so that tests to be complete should include the effects of wall thickness. They should also bring out the relative pressure or suction in the furnace as compared with the air on the outside.

Mr. Martin had spoken of steam-atomizing burners. As brought out in the paper, a steam-atomizing burner outfit was ordinarily simpler than a mechanically atomizing burner, and many were used today although, as a rule, in the smaller plants.

Mr. Martin had also spoken of a steam-atomizing burner using a register and a conical flame similar to the mechanically atomizing burner. These burners could be installed one above the other in the same way as the mechanical burner and thus develop a higher capacity than the more usual form of steam burner with a flat flame and air introduced from below. Burners of this sort had been developed by the Babcock & Wilcox Co., Ltd., and had been used to some extent by them.

Mr. Gibson had brought out the idea of buying the greatest number of heat units for the dollar, which was certainly something to be looked into.

The remarks by Professor Trinks on the glazing of the inside of the furnace to prevent the hot gases leaking outward or the air leaking inward to too great an extent were most interesting. As had often been stated, in the practice of building furnaces for boilers, the difference between a suction of a twentieth of an inch inside the furnace and a pressure of the same amount might mean the difference between the success and failure of the brickwork, and the action of the slag in preventing the passage of gas or air was an important feature. The brickwork of a narrow, high furnace would not stand up as well as the same brickwork in a wider furnace on account of the reflection of the heat from one wall to the other.

In connection with large and small furnaces used in oil burning mentioned by Mr. Smith and Commander Donald, the use of small furnaces was compulsory in marine practice as was brought out in the paper. In marine practice, distilled water was available for feedwater, and this as a rule was better than the feedwater of stationary plants. In running a small furnace to the limit of capacity, all the

[1] Ch. Engr., Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. Mem. A.S.M.E. Dr. Jacobus presented Mr. Lewis's paper in the absence of the author.

burners must be accurately adjusted and every detail watched with great care. On the whole, the tendency in stationary practice had been toward larger furnaces, and many of these furnaces had been installed and the boilers set so that a change could be made from oil to coal if necessary.

Regarding Mr. Gibson's remarks about the fallacy of the general impression that by changing a plant from coal burning to oil burning more trouble was experienced with the tubes, it was true, as Mr. Gibsons aid, that the trouble came from the boiler water rather than from the oil burner. Boilers might have scale in them in an amount which would cause no trouble when run at a certain capacity with coal fuel, but which might give trouble with oil fuel at a higher rating, especially where the furnace volumes were limited and the heat localized on portions of the tubes. As Mr. Gibson had pointed out, this was not inherently a trouble due to burning oil; with a proper oil-burning equipment it was simply due to the fact that the boilers were dirty inside and were not in a condition to operate at high capacity.

HAZARDS OF INDUSTRIAL OIL BURNING

E. A. Barrier[1] said that the flash point, as Mr. Newell had pointed out, had been the subject of controversy in the last two or three years, and the position had been taken by some that the figure of 150 deg. was arbitrarily chosen.

However, it represented the judgment of experienced fire-protection engineers who had taken everything into consideration, and therefore it was no more arbitrary than any factor of safety chosen in engineering practice.

Regarding untopped oils, he agreed entirely with the author that there was danger in the use of these oils. Even though crude oils had but three to five per cent of volatile constituents, that was ample to produce an explosive mixture inside the tank, as when oil was drawn from the tanks air came in, making an explosive mixture.

The boiling over of topped oils was probably due to a very small percentage of moisture which was held in colloidal suspension in the oil, and which turned into steam and caused serious foaming.

Five or six years ago some tests on siphoning were conducted in his company's laboratory. Siphoning was purely a question of the size of the pipe, the viscosity of the oil, and the difference in head. Their results, which were limited to one particular condition with a 2.5-in. pipe and an oil of 13 deg. B. heated to 110 to 120 deg. fahr., showed that it would siphon very slowly with a difference in head of about 10 ft. With less, there was practically no siphoning. If the oil were heated to a higher temperature, and if the pipe were larger, the siphoning would be easier.

In his opinion altogether too much emphasis had been laid on vapor-tight tanks. There was no question but what so-called vapor-tight tanks would leak after they had been in service a few years and had been subjected to the change of temperature and the warping and twisting that a structure of such size would have to withstand.

J. P. Leask[2] asked what method of guarding against corrosion had been most efficient.

Mr. Barrier in reply said that it was his experience that corrosion in pipes or tanks was due to local conditions which usually could be eliminated.

F. M. Gibson said that the connections at the end of metallic hose should be mechanically fastened rather than soldered.

The study of soot should be investigated further. Sparking would occur in the bottom of a stack in the flow of hot gases where the temperature was perhaps not over 125 deg., but if the same soot was taken into a laboratory and hot air blown over it it would not kindle below a temperature of 600 deg.

E. B. Powell said that in a central station where the entire personnel was trained in careful attention to details and the plant as a whole was practically fireproof, he could see no objection to the use of untopped crude oil. His experience with straight crudes of various gravities from 20 down to 12 or 10 had indicated no difficulty whatever in the hands of central-station crews.

H. L. Shoemaker[3] said that the various rules and regulations of the National Board of Fire Underwriters, and the National Fire Association, and other agencies had been subject to much discussion during the last two or three years. The oil interests had felt that many things might well be modified, basing their opinions on their own experience and on the views of their experts. It certainly seemed unfortunate that in several sections of the country large organizations had been unable to put in large oil-storage and handling equipment on account of municipal restrictions or the ideas of people which had no reasonable basis.

W. N. Polakov[1] mentioned the instance of a small industrial plant in a little town of Massachussetts that contemplated using crude oil and found that insurance companies or underwriters objected. Across the street from the plant, however, there was a garage with seven gasoline storage tanks.

Mr Newell, in closing, said that there was one hazard with crude oil which did not exist with gasoline, namely, that of boiling over. Gasoline did not act that way. When a tank containing gasoline was involved in a fire, the top went off, if it was one of the older tanks with the top so constructed that the rivet spacing was wide, and the gasoline burned like an open torch. Crude oil had all the light elements in it that were not ordinarily in gasoline, and in addition there was the hazard of boiling over. In a roofing plant near Elizabeth, N. J., there were two tanks containing what was called "flux," which was used as a basis for roofing material. The wooden tops of these tanks took fire, and the sparks from them spread to a tank with a wooden roof containing Mid-Continent crude oil. In the meantime the fire in the flux tank was extinguished and much of the contents of the tanks was salvaged; but the crude-oil tank boiled over three different times, and in boiling over it flowed beneath and completely surrounded a tank containing naphtha, which was probably a little lower fraction than gasoline. The naphtha tank was mounted on concrete piers and had on it a 6-in. safety valve. At the end of the fire the tank was intact and was still about two-thirds full of naphtha. On the other hand, the escaping crude oil flowed down an incline and surrounded a roofing plant which was practically new. The loss of the roofing plant amounted to about a million dollars. This was a good comparison of the hazards of gasoline and crude oil under fire conditions.

ALTHOUGH the so-called "short-time fatigue test" apparently will not replace the endurance-run test, sufficient correlation between the two types of test has been found to make further study worth while.

The controlling factors in the design of specimens are different for short-time "fatigue" tests and endurance-run fatigue tests. In endurance runs it is necessary to secure failure at a place where the stresses are determinate and calculable. The highest maximum of stress lies on a free portion of the specimen far enough from the clamps to be calculated by the elementary Euler-Bernoulli theory. The shape of the specimen must also provide for a maximum of stress sufficiently removed from clamps and fillets to be calculated by this simple theory and large enough to exceed any local stresses near the clamps.

In the short-time "fatigue" tests failure is not desired. There is, therefore, no necessity of providing a maximum of stress on a free portion of the surface of the specimen. In addition, the indeterminate stresses concentrated in the neighborhood of the clamps contribute only a small part to the deflection or temperature rise, so that there is no practical disadvantage in allowing them to be higher than those in the uniformly stressed free portion of the specimen. Those limitations which determine the design of endurance-run specimens, therefore, do not apply to short-time tests. It is possible, then, to design a specimen for this test with maximum stresses uniform over a large portion of the material, thus securing greater sensibility. This should be the controlling factor in the design of these specimens.

Specimen shapes are shown suitable for different types of short-time "fatigue" tests. Because of the simple specimen shape the Sondericker type of machine is deemed best suited for these tests. (L. S. Tuckerman and C. S. Aitchison (Mem. A.S.M.E.) in *Abstract of Technologic Papers of the Bureau of Standards*, no. 275, *d*)

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[1] Arkwright Mutual Fire Insurance Co., New York, N. Y.

[2] 2nd V.-P., Peabody Engrg. Corp., 110 E. 42nd St., New York, N. Y. Mem. A.S.M.E.

[3] Standard Oil Co. of N. J., Elizabeth, N. J.

[1] Pres., Walter N. Poakov Co., Inc., 25 Fifth Ave., New York, N. Y. Mem. A.S.M.E.

Hardness Testers

Description of a New Hardness-Testing Machine Operating on the Brinell Principle—Discussion of Professor Keller's Paper on the Herbert Pendulum and other Hardness Testers—Coker's Research Work on Cutting Metals

AT THE recent Annual Meeting of the A.S.M.E., a session was devoted to research in machine design and operation under the auspices of the Research Committee on Cutting and Forming of Metals of the Machine-Shop Practice Division, B. H. Blood,[1] Chairman of the Committee, presiding. J. O. Keller,[2] presented a paper[3] entitled Comparison of Herbert Pendulum Hardness Tester with Other Hardness Testers, in which he described the Herbert hardness tester and a set of tests made with it and other testers by which comparative results had been obtained and analyzed. Joseph K. Wood,[4] Chairman of the Special Research Committee on Metal Springs, also presented a preliminary progress report[5] of the Committee.

Orlan W. Boston[6] made a brief report on research work in the cutting of metals. He explained that the Committee was trying to determine elementary or fundamental equations for the cutting of metals by straight-line motion. This equation was to be a function of the geometric form of the tool, the quality of the material being cut, the quality of the tool, the speed, etc. He said that already there were numerous conflicting statements on the subject which he hoped to cover in a more detailed report. In England he had visited the laboratories of Professor Coker at the University College of London. Professor Coker was a member of what corresponded to the Society's Committee on Cutting and Forming of Metals. He had done some very creditable work in cutting nitrocellulose with a glass tool, examining the material being cut as well as the tool with polarized light, and determining the stresses. He claimed that the results of his experiments were applicable to steel and cast iron because these metals, like nitrocellulose, had uniform values of heating below the elastic limit. This, however, was a question yet to be proved, because it was known that the formation of a chip was a result of stress beyond the elastic limit, and whether steel and nitrocellulose were similar beyond the elastic limit was still to be determined.

Discussion of Paper on Herbert Hardness Tester

The discussion of the paper by Professor Keller was opened by Professor Boston who gave particulars regarding a new hardness-testing machine developed and recently put on the market by Messrs. Vickers, Ltd., Vickers House, Broadway, London, S. W. 1. Professor Boston had enjoyed a demonstration by Mr. Sandland, the designer, at the British Empire Exhibition and had been very much impressed by its commercial possibilities.

The machine, shown in Fig. 1, was a self-contained unit, being automatically driven, and yielded Brinell hardness numerals of a thoroughly reliable order free from the irregularities to which the Brinell test was exposed by the personal equation of the operator.

A description of the Vickers machine had been given in *Engineering*, April 25, 1924. The company, in indicating the advantages of the machine, accounted for the irregularities in the usual Brinell reading as follows:

1 Inertia and momentum caused by lifting the load and applying it too rapidly. (The effect of time and speed, together with definitions of hardness, were discussed in the Symposium of Hardness Testing, published in Trans. A.S.S.T., February, 1923, p. 480.)

2 Variable duration of the load.

3 Deformation of the steel ball or indenter. This, of course, was most evident on hard materials, it being generally agreed that the figures were unreliable above 600, although some authorities put this at an even lower figure.

4 Difference in deformation of the materials tested. On a given homogeneous material different numerals resulted from different loads, i.e., different amounts of deformation, due to geometrical dissimilarity between ball impressions of different sizes.

5 Errors in measurement.

In the machine described, provision was made for the elimination of these irregularities as follows:

1 and 2 The load was lowered on to the specimen, being applied very slowly and at a diminishing rate. This application and the removal after a predetermined interval were controlled entirely automatically. Inertia and variable duration of the load were thus eliminated.

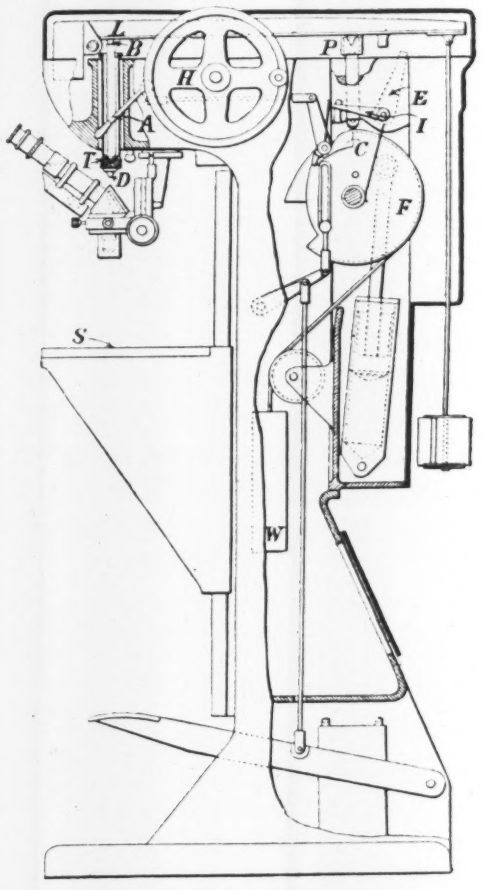

FIG. 1 DIAGRAM OF THE VICKERS HARDNESS TESTER

3 The indenter was a diamond, which was incomparably harder than any steel or other metal. It followed, therefore, that deformation of the indenter would be negligible.

4 The diamond was cut to the form of a square-based pyramid. It would be seen that this must yield impressions which were geometrically similar and must therefore, within wide limits of size, give comparable results. A similar effect was, of course, obtainable with a cone, but on certain materials it would be found that all circular impressions were both difficult and uncertain to read owing to the lack of distinctness which occurred at the extreme edge of the impressions. This difficulty did not arise when using a pyramid, the impressions of which were measured across the diagonal corners.

It had been generally advocated that all ball impressions should, if possible, be within the range 0.25 to 0.5 of the diameter of the ball, and, accepting the mean of these (i.e., 0.375) as the ideal, the equivalent angularity of impression had been taken as standard for the pyramid.

It would be seen from Fig. 2 that the angle was 136 deg. and it had been found that the hardness numerals obtained with such a pyramid were equivalent to those resulting from ball impressions of the ideal size. Hardness was calculated as load divided by pyramidal area.

Tests had shown that, for a given sample, constant hardness numerals were obtained with the Vickers machine for different

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[1] Gen. Mgr., Pratt & Whitney Co., Hartford, Conn. Mem. A.S.M.E.

[2] Professor of Industrial Engineering, Penn. State College, State College, Pa. Jun. A.S.M.E.

[3] See MECHANICAL ENGINEERING, Mid-November issue, vol. 46, no. 11a, 1924, p. 818.

[4] Cons. Engr., New York, N. Y. Assoc-Mem. A.S.M.E.

[5] See MECHANICAL ENGINEERING, Mid-November issue, vol. 46, no. 11a, 1924, p. 793.

[6] Asst. Prof. Shop Practice, Acting Dir. of Shops, Univ. of Mich., Ann Arbor, Mich. Mem. A.S.M.E.

loads or impressions, while with the Brinell machine increasingly reduced numerals were obtained for impressions below 0.375 of the ball diameter and increased numerals for impressions above 0.375 of the ball diameter. The Brinell reading for an impression 0.375 of the ball diameter corresponded with the constant reading of the Vickers machine.

5 The measurements of the impressions were taken by means of a specially constructed micrometer ocular, which was fitted with knife edges instead of the usual hair lines, thus reducing eye strain. The readings were taken entirely from actual figures shown at the side of the ocular, so rendering impossible any errors due to the miscounting of divisions on a small scale.

The impression appeared as a dark square on a light ground, and measurements were taken between knife edges in the ocular across the diagonals of the square as shown in Fig. 3.

It would be seen from Fig. 1 that the machine was of the simple-lever type, a pivoted loading lever L applying the load through a

FIG. 2 RELATION OF BRINELL BALL TO VICKERS DIAMOND INDENTER

FIG. 3 KNIFE EDGES ADJUSTED TO MEASURE THE DIAGONAL OF THE SQUARE IMPRESSION

cone B and thrust rod to a tubular guide and holder T for the indenter D.

At the beginning of the test the load lever was supported by the cam-controlled plunger P, the vertical reciprocation of which therefore applied and removed the load. In order to carry out a test, the microscope was swung out of position and the specimen was placed upon the stage S, which might be raised by rotation of the hand wheel H, operating through bevel wheels and a screw; by this means the specimen was adjusted to be just clear of the diamond.

The cam C, controlling the plunger, was mounted on a drum F and when the starting handle A was pressed, both were rotated by the motor weight W. The movement of the cam at first lowered the plunger, thus applying the test load to the specimen; further rotation raised the plunger and removed the load, the whole operation being completed in 90 deg. rotation of the cam. The speed of rotation was determined by a piston and oil dashpot, the rate of the displacement of the oil being regulated by an adjustable control valve. In the upper end of the plunger was a rubber pad which, engaging with a cone mounted in the beam, insured a very slow and diminishing rate of application of the last portion of the load. The valve was set so that 30 sec. were used in the application and duration of the load, and the cam was so designed that, of this period, the application occupied 20 sec. and the duration a minimum of 10 sec.

Visible and aural indication was given at the end of the test by the indicator I, which clicked into the "finish" position. Depression of the foot pedal returned the cam, drum, weight, and piston to their original positions, this movement being facilitated by the opening of a non-return valve in the piston. A tripping piece E meantime supported the beam and dropped out as soon as the machine had been reloaded.

The measuring microscope was of the angular type, mounted on a hinged bracket which was locked into the correct position by a catch. The illumination was vertical, and was obtained from lamp and condenser mounted on the microscope itself. Accumulators in the base of the machine provided the necessary current for the lamp.

The diamond might be replaced by a steel ball and the machine used as an extremely accurate form of Brinell ball tester, the load applied was very light (standard load 50 kg., maximum 100 and minimum 10), and the impression was very small. The machine was therefore applicable to almost any class of work, such as case-hardened steel, thin sheet steel, small tools, small areas, etc. The machine had a capacity up to 13 in. in depth. Some advantages claimed by the Vickers Company for this machine might be summarized as follows:

1 The figures obtained were Brinell. These were the most widely accepted and recognized in the world.

2 The Brinell figures obtained were always reliable. This was not possible with any other machine.

3 Uniformity of load. No inertia, since the load was automatically applied in a scientific manner

4 Uniformity of duration of load. The load was removed automatically at the end of the predetermined interval.

5 No limit to hardness of metals that could be tested accurately, the diamond being many times harder than the hardest steel.

6 No damage to finished work, the impressions being very small.

7 Applicable to either the "case" or "core" of case-hardened work. The impressions were shallow, about 0.002 in. deep on hard steel.

8 No resetting job under the microscope, the latter swinging into the correct position over the impression.

9 No errors in reading. The readings were taken from actual figures instead of a scale.

10 Rapid checking to specified limits. Knife edges in eyepiece provided plus and minus limits.

11 Application to large and small pieces. Work up to 13 in. deep could be accomodated, and in special cases this might be increased up to 17 in.

The next discusser, A. V. De Forest,[1] stated that the careful comparison of the various types of hardness tests made by Professor Keller in his paper offered an excellent opportunity for correlation with magnetic tests. In order to indicate the relationship established so far, the speaker exhibited a chart on which he had plotted the principal measurements for comparison. This chart showed curves of Brinell, Herbert time test, Herbert scale test, scleroscope, Rockwell, and one magnetic test so arranged that if the results were proportional, the curves would be parallel with each other Inspection showed that while all tests placed the hardness of specimens C, B, and E in approximately uniform steps, they did not agree so well on AA and particularly on F. In this latter case the scale test and the Rockwell and magnetic tests indicated that a factor was in evidence that differed from those affecting the other tests. If measurements on a large number of specimens confirmed these results in the future, it would seem that the relation between Brinell and Rockwell hardness might afford a distinction similar to that indicated by the relation between time test and scale test on the Herbert pendulum. The whole matter of interpretation in terms of machinability would of necessity await the results of direct tests of this property.

On another chart presented by Mr. De Forest, the results of two different magnetic measurements were plotted against each other. One coördinate represented the hysteresis loss and the other permeability. These magnetic properties were to a considerable extent independent of each other and depended on different characteristics of the material. It was interesting to note that specimen F was so far out of the family group. If this was due to the fact that this material was of large grain size and contained a cementitic network, the magnetic test might be of service in indicating this undesirable condition.

Both measurements were made on the same galvanometer and plotted directly on the record paper without the use of any numerical scale. In practice the specimen was placed in a solenoid and, by throwing a switch, either the X circuit, measuring loss, or the Y circuit, for permeability, could be connected with the galvanometer. The spot of light indicating the reading was thrown on the under side of the record paper, and could be marked by a pencil point. If further work substantiated out the present indications that specimens F and A should be grouped together as coarse-grained pearlite structures, the constants of the apparatus could easily be so adjusted that F and A would lie along a line parallel to the X-axis. A measurement on the Y-axis would then distinguish these from the other specimens regardless of the actual difference in hardness between the two.

In this remarkable flexibility, according to Mr. De Forest, lay the hope that magnetic methods of test and inspection would provide the tool to improve greatly the quality of all our steel products.

[1] Am. Chain Co., Bridgeport, Conn.

A. H. d'Arcambal[1] said that Professor Keller's paper was valuable because it was the first that gave results obtained with the Brinell, scleroscope, Rockwell and Herbert hardness testers on the same specimens of carbon tool steel.

Mr. d'Arcambal found that the Herbert pendulum hardness tester was more or less of a laboratory instrument, but trusted that it would be redesigned so as to make it suitable for production work. He did not believe that this instrument indicated the machining qualities of steels to any greater extent than did the other types of hardness testers now on the market, especially as regarded alloys and high-speed steels. The machinability of plain carbon tool steel was, he said, fairly well determined by taking the scleroscope, Rockwell, or Brinell reading, although at times the microstructure played an important role. Two bars of carbon tool steel might possess the same hardness number but machine differently due to a difference in the microstructure of the material in question.

It was well to remember that in speaking of machinability, such machining operations as turning, drilling, reaming, milling, planing, threading, etc. had all to be included. A piece of tool steel should be fully annealed if the best results were to be obtained in turning in automatics, but this steel would be a little too soft for milling and threading and altogether too soft for planing. High-speed steel annealed to a Brinell hardness of 228 was in excellent condition for turning, but a Brinell hardness of at least 400 was necessary to obtain a smooth surface after planing. It would be commercially impossible to turn, mill, or thread high-speed steel possessing a Brinell hardness of 400.

The Pratt & Whitney Co. had found that the Rockwell hardness tester with the diamond cone was the most satisfactory hardness tester for finished tools. The Brinell instrument at times gave incorrect results due to the work being tested possessing a greater hardness than the 10-mm. ball cover. It was also difficult to read accurately the diameter of an impression 2.5 mm. or smaller. Moreover a load of 3000 kg. exerted on hard material set up severe strains in it, and the scleroscope required the specimen under test to have fairly well-polished and parallel surfaces if the specimen was clamped to the standard anvil. Samples held in a vise and tested for scleroscope hardness usually showed a lower reading than they really possessed.

The Rockwell instrument was not as rapid a tester as the scleroscope but was considerably more rapid than the Brinell. The advantage gained by using the diamond cone for testing hard material was obvious. Moreover highly polished or absolutely parallel surfaces were not required for accurate readings. The fact that the Rockwell tester gave the reading directly on the dial was of considerable advantage as compared with the necessity of having to read the diameter of the impression produced by the Brinell ball. Mr. d'Arcambal stated that his company had recently obtained the Rockwell hardness numbers on hardened carbon tool steel, hardened high-speed steel, and No. 2 stellite at temperatures varying from room temperature to 1200 deg. fahr., and found the Rockwell tester to function satisfactorily when testing the hardness of metals at elevated temperatures.

Stanley P. Rockwell[2] said that Professor Keller's idea in trying to correlate the various machining properties of metals with existing forms of hardness tests would, when completed, undoubtedly throw considerable light on a subject hitherto only guessed at. He had had the pleasure of lending some minor assistance to Professor Keller during his initial work and regretted that he had not had the opportunity of reviewing certain of his results as they pertained to the Rockwell hardness tests. Had he been able to do this he could have corrected an erroneous impression given by him to Professor Keller that tests on a scale-covered surface were a reliable indication of interior or true metal conditions. As Professor Keller had averaged scale-surface readings and scale-removed readings, it was apparent that Table 1 was in error and conflicted with results obtained by Dr. Cowdry, Mr. Spaulding, and others, especially as regarded Rockwell-Brinell relationships.

As Professor Keller had revised his Rockwell data and used results made only on scale-removed surfaces, his Table 1 became consistent with the different forms of test.

In regard to the Herbert tester's indicating machinability, Mr. Rockwell trusted that in Professor Keller's subsequent work he would define "machinability." He was under the impression that a steel which would turn freely might not drill easily. On the other hand it might mill readily, but clean tapping might cause difficulty. Moreover these conditions might be altered through tool, rake, clearance, speed, or lubrication.

Further, he would like to suggest to Professor Keller that he investigated the a and n values as they affected the so-called machinability as so ably discussed by Dr. Hoyt at the Boston A.S.S.T. symposium on hardness testing. These might be obtained on the Rockwell as well as the Brinell direct, using the relationship $P = at^n$, where P was the load in kg., t the reciprocal of the Rockwell number, and a and n were the constants to be determined.

In answer to a question by Mr. de Forest in regard to determining the hardness at the bottom of the Brinell indentation, Professor Keller said that he had carried out such a test but remembered only that the material was harder at this point. In the case of the Herbert pendulum, he pointed out, the hardness increased slowly. As Mr. Herbert had indicated, some materials were hardened immediately, that was, after the first oscillation of the pendulum, the hardness reaching a maximum after one or two swings of the instrument, while on the other metals the instrument might swing ten or twelve times before maximum hardness was reached. Undoubtedly all materials were harder after a Brinell test, but some metals worked up slowly and others immediately to the maximum hardness, and Mr. Herbert believed that the former would draw better.

C. A. Beckett,[1] referring to the relation between the pendulum tester and the cutting that was done by Dempster Smith, said that the principal advantage the latter found by subsequent cuttings and by comparing the results after each cut with the Herbert hardness tester was in trying to determine a measure of machinability. The formula he used and the results obtained checked closely with Brinell. It should be noted, however, that Smith used but one kind of steel, and that he did not change the tool. For this reason the experiments were limited in scope.

Chairman Blood said that it was evident that in the new field which was being opened up there was much work to be done. The inventor of the Herbert method of testing claimed certain advantages over better-known instruments. One, for instance, was in testing very thin sections of metal. The impression made by the Herbert ball was very shallow and was not much affected, perhaps not at all, by thicknesses even as small as a razor blade. Another advantage was in testing very hard pieces. Mr. Herbert had told him that he could distinguish between pieces of high-speed steel with identical hardening, one of which had been drawn to 1050, another to 1075 and a third to 1100 deg. fahr. The Committee's preliminary experiments had not proven this to be true, but the inventor's statement was none the less interesting if it could be proven.

The instrument had the disadvantage of being hardly rugged enough for shop use, although when mounted in the stand now provided by the manufacturers it became a more practical instrument to put into the hands of an inspector on production work.

Louis H. Chasan,[2] stated that he did not know of any instrument that could determine the hardness of sheet metal for drawing work as well as the Erichsen sheet-metal tester. He had found it to be a very reliable instrument.

Professor Keller, in closing, said that he was much interested in Professor Boston's description of the new Vickers hardness tester, especially in its advantages over the usual method of securing consistent Brinell numbers. Mr. De Forest had also brought forth some valuable information and had promised to make further tests in connection with the work that was expected to be carried on in the future. It was possible that in further tests, samples would be so chosen as to facilitate the work of magnetic testing. He would also give Mr. Rockwell's suggestion concerning the a and n values discussed by Dr. Hoyt at the Boston A.S.S.T. meeting some attention in other work. In work on the drawing properties of metals, an Erichsen sheet-metal tester would be secured to compare with the Herbert method of work hardening. By the time these future tests were completed, there might be material secured of a nature that would make it worth while to publish a second paper covering the same subject.

[1] Metallurgist, Pratt & Whitney Co., Hartford, Conn.
[2] 61 Highland St., Hartford, Conn. Mem. A.S.M.E.

[1] Columbia University, New York, N. Y.
[2] Singer Mfg. Co., New York, N. Y. Jun. A.S.M.E.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

High-Specific-Speed Hydraulic Turbines

A SYMPOSIUM of six papers on high-specific-speed hydraulic turbines, in particular in their bearing on the proportioning of the number of units in low-head hydroelectric plants, was presented at the session of the Power Division of the American Society of Civil Engineers, held at the annual meeting of the society in New York City on January 22, 1925.

Two of these papers, one by Geo. A. Orrok (Mem. A.S.M.E.) and the other by Charles C. Egbert (Mem. A.S.M.E.), are mainly descriptive of present installations. A paper by Geo. A. Jessop compares the efficiencies of medium-speed wheels with those of high speed wheels. Two papers, by Lewis F. Moody (Mem. A.S.M.E.) and Forrest Nagler (Mem. A.S.M.E.), respectively, go into questions of design of the turbines themselves, while a paper by John P. Hogan discusses the important matter of proportioning units for a hydraulic electric plant.

The key to the whole subject is given by the introductory paragraph of the paper by Lewis F. Moody: "It must seem surprising to any one who has followed the development of engineering in the hydraulic-power field that a machine so highly developed as was the hydraulic turbine ten or fifteen years ago, should have suddenly entered on a period of radical changes and encountered transformations in many of its essentials. The fact remains, however, that although practice in turbine design had become almost standardized in the period just prior to the World War, the development of this art has taken on new life and has already been revolutionized in many aspects."

Development of the High-Specific-Speed Turbine Abroad

The development of the high-specific-speed turbine abroad is well indicated by the data presented in Geo. A. Orrok's paper, from which Table 1 is reproduced. As stated by Mr. Orrok, Dr. Thoma in his paper before the World Power Conference emphasizes the fact that the tests on small-sized, geometrically similar models give reliable figures for large-sized operations, and that in the few cases of divergence noted a lack of geometrical similarity has been proven. He also states that the Camerer correction formula for efficiency has been found to be fairly correct. He gives a list of thirteen hydraulic testing laboratories actually running in Germany alone.

TABLE 1 EUROPEAN LOW-HEAD TURBINES

Manufacturer	Location	Diameter ft.	Diameter in.	Discharge in cu. ft. per sec.	Head in ft.	Revolutions per min.	Horsepower	Specific speed
Charmilles, S. A...	Chauncy-Pougny, Switzerland	18	0	3250	26.7	83.3	7600	120
A. B. Finshyttan[1].	Lilla Edet, Sweden	19	8	5200	21.3	62.5	10000	136
A. B. Finshyttan[2].	Lilla Edet, Sweden	19	0	5300	21.3	62.5	11200	140
Voith.....	Seibenbrunn, Austria	6	3	...	20.3	250	400	212
A. B. Karlstads...	Gavle, Sweden	5	1	...	13.75	250	400	219
Voith.....	Kachlet, Germany	15	1	...	25.1	...	7450	115
Neumeyer.....	Viereth, Germany	9	6	...	17.4	...	2000	93
Vevey.....	Wynau, Switzerland	10	10	1700	17.0	107	2700	165

[1] Lawaczeck wheel. [2] Kaplan wheel.

Additional information on the same subject is supplied in the paper, Some Applications of the Propeller-Type Water Turbine in Europe, by Charles C. Egbert. Of interest are the details given regarding the propeller-type turbine in the Matte central station of Berne, Switzerland, made by Theo. Bell & Co. of Kriens, Switzerland. In this small installation the runner is of the two-bladed screw type, each blade covering about 180 deg. of the circumference. The turbine was installed under the following guarantees for operation under a head of 11.25 ft. (3.45 m.):

Speed in r.p.m.	200	250
Output in horsepower	290	323
Efficiency in per cent	80	78

from which the specific speed may be computed as about 165 ft-lb. units (725 metric) at 200 r.p.m. and 216 ft-lb. units (955 metric) at 250 r.p.m.

One of the latest installations is that of the Wynau Electricity Works on the Aare River in Switzerland, which includes two units of 2700 hp. nominal capacity each, operating at 107 r.p.m. under a variable head of from 8.2 ft. (2.5 m.) to 17 ft. (5.2 m.) and supplied by the Ateliers de Construction Mécaniques de Vevey.

Acceptance tests of these turbines indicate a maximum efficiency of 89 per cent, with a specific speed of 165 ft-lb. units (728 metric) when operating under 17 ft. head (5.2 m.). Under 8.2 ft. (2.5 m.) head at 85 per cent efficiency the specific speed is given as 195 ft-lb. units (862 metric).

Design of Propeller-Type Turbines

According to Lewis F. Moody, the most striking changes made in recent designs are those in runners and draft tubes, resulting in a remarkable increase in the specific speeds. His paper is devoted particularly to one form of high-speed turbine. Among the subjects he discusses are variation of efficiency with the size of the turbines. He does not consider the Camerer formula satisfactory and develops on the basis of the latest information a formula of his own, namely,

$$e' = 1 - (1 - e)\left(\frac{D}{D'}\right)^n$$

where D and D' are runner diameters of the turbines, and e and e' the respective efficiencies of the turbines. A number of actual tests have shown that n varies but little from the value of 0.25. This formula is applicable, however, only to geometrically similar turbines, and Mr. Moody points out the principal alteration which a change of form or type of turbine involves, in particular, a change from the so-called Francis turbine to a turbine of the propeller type. The effect of the changes which he enumerates is to allow the runner to be operated at a higher speed under a given head by permitting higher relative velocities to be used and a more inclined direction of the absolute velocity of runner discharge.

Among other alterations in passing from the older to the more modern type is the introduction of a large transition space between guide vanes and runner, in which the water follows a free path and turns from a radial to a partly or a completely axial direction.

The effect of the transition space is to reduce the relative velocity between the water and the blades of all sections except at the blade tips. If this space is too much enlarged, however, or if the entrance edges of the blades are carried too near the axis, the relative velocity near the hub begins to increase again, the water at that point then tending to rotate faster than the runner, and, in addition, the surface friction due to the long absolute path along the stationary head cover partly discounts the decrease in blade friction. This consideration is one of those which has led to Mr. Moody's diagonal type of runner, intermediate between a radial- and a purely axial-flow runner. In a similar way, an undue reduction in blade surface while decreasing the surface resistance, if carried too far results in

imperfect guidance of the flow, and permits other losses which more than offset the gain. It has been found both from theory and experience that unduly small blade area greatly impairs the stability of operation, giving rise to surging, unstable flow, and loss of efficiency, and materially reduces the field of application of turbines of the propeller type in which it is found.

Mr. Moody derives formulas showing that in such a series of runners as he describes the blade friction will vary directly as the cube root of the number of blades, and inversely as the cube root of the radius. Because of this a reduction in the number of blades is of advantage, provided it is not carried too far.

After showing by curves the efficiencies plotted against specific speed for turbines of propeller type of the form developed by him from Holyoke tests, Mr. Moody describes a few recent installations such as The LaGabelle plant, and the Feeder Dam on the Hudson River near Glens Falls, New York. The former are similar to the 28,000-hp. turbines of the Manitoba Power Co., Ltd. It may be reported as a result of the experience with these units to date that the Manitoba runners have shown a slight tendency toward corrosion on the discharge edges of the blades, due perhaps to the abnormal operating heads, but the effect is local and not serious. The LaGabelle runners are in perfect condition. In both plants the concrete just below the runner has been eroded, and this has been corrected by the addition of plate-steel liners at the top of the draft tube, a practice which will be followed in future installations of the propeller type under the higher heads.

At the Feeder Dam plant there are five units with a rated capacity of 1500 hp. each, operating under a normal head of 15.5 ft. and a maximum head of 19 ft. The speed is 120 r.p.m., corresponding to a specific speed of 151.5 (672 metric). The design has been simplified as far as possible and the object has been to keep the cost of turbines as low as possible without sacrificing mechanical dependability or the hydraulic conditions of operation. On account of the low head available for this station, a minimum cost of the development is naturally of great importance. In this plant an outside operating mechanism has been dispensed with, the gates being operated from links connected directly to the vanes, by means of a vertical gate shaft. The runners are of cast iron, the casing stay vanes are separate castings set individually in the concrete, and no continuous "speed ring" is used. The draft tube is a relatively short spreading tube with the barrel or spreading portion of plate steel. These units have been in actual operation for several months and tests of power output have shown excellent agreement with the predicted values.

In installations now being made in Canada of two turbines of 6000 hp. unit capacity for heads of 30 ft. normal and 34 ft. maximum, the draft tubes are of the spreading type, formed in concrete, and will be provided with separate cast stay vanes at the end of the spreading portion to carry the weight of the superposed structure. Similar stay vanes were used in the Manitoba and LaGabelle installations and in a large number of the Francis turbine installations. These vanes have been found to be a valuable structural feature, reducing the reinforcement in the power-house substructure which would otherwise have to carry the loads by beam action, and also proving useful during construction as they permit the earlier removal of forms.

The five units the construction of which is just being completed for the Illinois Northern Utilities Co. are of particular interest because they are to have a unit capacity of 800 hp. under a head of only 8 ft. operating at a speed of 80 r.p.m. Here every effort has been made to keep the cost of installation at a minimum. Separate stay vanes have been used, bolted at the top to the foundation ring. This foundation ring is built up of plates and structural shapes and is continuous with the plate-steel pit liner carrying the operating cylinders.

As a result of experience with this new form of turbine certain minor difficulties developed which required for their correction some modifications in design, but the many major difficulties which were predicted did not occur.

Bearing of High-Specific-Speed Turbines on the Proportioning of the Number of Units in Low-Head Plants

In his paper entitled, High-Specific-Speed Hydraulic Turbines in Their Bearing on the Proportioning of the Number of Units in Low-Head Hydroelectric Plants, Geo. A. Jessop endeavors by means of curves to indicate the comparative efficiencies which can be obtained by using medium-speed wheels with a specific speed of about 75 and high-speed wheels with a specific speed of about 175. The efficiencies shown throughout this paper are based on model runners of the size usually tested at the Holyoke flume—about 39 cu. ft. per sec. discharge at 1 ft. head. It is possible to obtain better results in place with larger wheels under good conditions. The data and information given in the paper were obtained from tests made in the Holyoke flume and in the specially designed and constructed testing flume of the S. Morgan Smith Co., York, Pa.

The medium-speed runner is of the Francis type and of normal design. The high-speed runner is a design first developed and used by Theodore Bell & Co. of Switzerland, and now being further developed and improved jointly by the engineering staffs of the Bell company and the S. Morgan Smith Company. It is distinctive in that it has fewer blades than is usual on high-speed wheels, and the total blade area is exceptionally large. The hub is especially large, thus reducing the usual amount of overlang of the buckets and providing a large cross-sectional area where each bucket joins the hub. An inspection of the curves shows that this wheel has its peak efficiency at about 90 per cent load, comparing very favorably with the slower-speed runner in this respect. This is a very desirable feature, because the turbine can be operated at its most economical point with a liberal allowance of power for regulating purposes, and furthermore, this design assists in obtaining high part-load efficiency.

On very low heads, i.e., heads of 30 to 40 ft. and under, the comparison discovered is possibly somewhat less favorable to the high-speed wheel than it should be, because of the conditions of test. The main purpose of the author is, however, to contrast the results that can be obtained with plants containing from two to eight units of the medium- and high-speed types. He begins by comparisons between two unit plants. He finds that for the medium-speed wheels when operating between 50 and 75 per cent load, higher efficiency can be obtained by running one wheel at maximum efficiency and supplying the rest of the demanded power by running the second unit at the required gate opening, than by dividing the load equally between the two units. The high-speed wheels show maximum efficiency when the load is equally divided. In order to determine whether it is economical to divide the load equally or unequally between two or more units, it is necessary to figure each plant separately, using the curves which apply.

This is done by using curves showing average weighted efficiencies. The "weighted efficiency" for a given range of load is simply the total horsepower output divided by the total water-horsepower-hours input, the latter being actually computed from discharge curves. In computing the average weighted efficiency for any given range of output, it is obviously necessary to use a definite load curve, and for purposes of comparison only Mr. Jessop has used a load curve which varies from the minimum to the maximum output at a constant rate.

For example, if the load varies from 40 to 100 per cent, the average weighted efficiency is about 89.5 per cent for the 75 N_s wheel and about 84.5 per cent for the 175 N_s wheel. If the load varies from 0 to 100 per cent, the average weighted efficiency is about 87 per cent for the 75 N_s wheel and about 80 per cent for the 175 N_s wheel.

For certain conditions Mr. Jessop recommends using the more easily determined arithmetical average rather than the weighted average.

The paper does not include a comparison of the average efficiencies under variable-head conditions, but one of the figures (Fig. 7 in the original) would indicate that the high-speed wheel has a better range at variable speed or variable head than a medium-speed turbine, and holds up well under low-head conditions. Hence where the percentage of head variation is considerable the comparison between the two wheels will be somewhat more favorable to the high-speed wheels.

Among other things, Mr. Jessop shows a comparison between European and Holyoke tests, from which it would appear that when the same runner, draft tube, and wheel case are used under the same head the difference between the tests is at no place equal

to 1 per cent, the comparison being made on the basis of the efficiency plotted against the specific speed.

HIGH-SPEED SUCTION TURBINES

Forrest Nagler, in a paper entitled, High-Speed Suction Turbines, outlines in a general way the experience to date with the high-speed axial-flow type of hydraulic turbine runner for low heads.

The high-speed, axial-flow type of hydraulic-turbine runner, called variously in the United States the "suction runner" or "propeller runner," is generally representative of a type wherein the flow within the runner itself is in a direction mainly parallel to or "cylindrical" about the axis of rotation. The particular type for which Mr. Nagler has been responsible is further characterized by a small number of blades (usually and preferably four) as contrasted with the 15 to 30 prevalent in present conventional forms, and by the absence of a rim around the tops of the blades. In addition there is a reduction in the width of the blade so that for the higher-speed conditions there is no overlap, this being essential to highest efficiency when high characteristic speeds are reached. It is with this type thus defined that his paper has to deal.

Practically all installations have followed the general practice conventional for the Francis type of runners as concerns a guide-case flume or spiral casing. On the discharge side practically every installation has made use of the White hydraucone regainer (see Trans. A.S.M.E., vol. 43, 1921, p. 255).

This type of runner has a lower efficiency on partial loads as contrasted with those of lower speed such as the Francis type; but while Francis runners of low capacity for low specific speeds have reached efficiencies as high as 90 per cent at half-load, they are not suitable for the application to low heads. The question of varying the capacity of the high-speed type of runner without materially affecting its efficiency is simply one of flattening the blade angles. In European practice the application of this principle has been carried so far as to make the blades movable under control of the governor, either simultaneously with or separately from the similar control of the guide vane.

Mr. Nagler does not believe that such an arrangement would be popular under American standards of ruggedness and because of the high cost of labor, and states that the need for such construction is not great where there are many units in the station. Governors are becoming less and less a factor, and he believes they will ultimately cease to enter into hydraulic-turbine design to any appreciable extent, their ultimate elimination being due to the fact that hydroelectric systems will be largely interconnected with steam power, where regulation is inherently less harmful to structures and to economy. He believes, however, that this feature of variable capacity will be utilized, not so much for its advantages in increasing part-load efficiency, as for the possibilities it presents in increasing the capacity under conditions of flood or low head. This feature has been utilized in numerous installations already built, construction details being used corresponding to the adjustment of pitch on screw pumps, which has long been regular practice in the United States.

It has been a fairly well-defined policy of the Allis-Chalmers Manufacturing Co. not to apply the suction type of runner to heads materially in excess of 30 ft., this being an arbitrary limit, although it is supported by experimental data as well as by theoretical considerations. By departing from the pure suction type of runner with its minimum blade surface and by using blades of larger projected area, the limiting head can undoubtedly be increased, but it is Mr. Nagler's belief and experience that the considerable increase of projected area, either by reason of wider blades or a larger number of blades, is accompanied by a very decided reduction of efficiency. It will undoubtedly be found that for certain installations a definite gain in some other direction may counterbalance this particular detrimental feature, and probably, as experience grows, this general type will be applied to higher and higher heads and to larger and larger capacities. At present, it is not believed that it is commercially advisable or sound from an engineering standpoint to apply it to heads of capacities far outside the range of proven practice. A consistent adherence to this policy has enabled the achievement of an operating record wherein there has been no single shutdown of a unit of as much as 24 hr. duration, due to conditions traceable to this type of runner.

Governor regulation has not been as perfect as that obtained with the corresponding type of Francis units, but this is principally attributable to the fact that at the higher speeds in question the factor WR^2 of the generator has been relatively lower. The regulation experienced has not differed from that obtained with Francis units by an amount unexplainable by this reduction of regulating constant.

The high-speed suction type of runner, due to its higher and sometimes critical velocities, is not conducive to as quiet a running machine as the Francis runner, perhaps in exactly the same way that the flow conditions in a Francis runner are not conducive to as smooth operation as is possible with an overshot wheel. Nothing, however, of a nature that would make it uncommercial has been experienced with this type when it is used under conditions to which it is inherently adapted, namely, relatively low head and large capacity.

PROPORTIONING UNITS IN LOW-HEAD PLANTS

A somewhat brief paper by John P. Hogan deals with the proportioning of units in low-head plants. Mr. Hogan enumerates the factors controlling in determining the number of units, assuming load regulation be provided from other sources. After considering the respective fields properly belonging to the Francis and high-speed type of runners, he comes to the conclusion that in peak-capacity plants it will be easier to arrive at a compromise in the number of units which at the same time will permit a large wheel and efficiency at relatively small daily flows, since it will be possible to operate for a relatively short period of time on a concentrated flow.

Other characteristics of the propeller type which should be considered are that they give better efficiencies at fluctuating heads and that they must be set much lower than wheels of the Francis type.

This leaves only the velocity of approach in the penstock and velocity of discharge to the tailrace to be considered. The limitations imposed by the propeller type are very similar to those of the Francis type, and in low-head plants of large flow these limitations are already presenting serious problems in the design and construction of waterways.

Already, under some conditions, the provisions of adequate water passages are of more influence in limiting the size of the units than the characteristics of the wheels, and with increasing use of wheels of the propeller type in large units, the influence of these hydraulic limitations will likewise increase. They may even be finally responsible for placing a definite limit to the increase in size of units for plants of large flow and relatively low head.

Short Abstracts of the Month

AERONAUTICAL ENGINEERING

Superchargers

IN 1924 the General Electric Company developed a new type of supercharger known as the "Side Type" for 20,000-ft. altitude operation. This device is located at the side of the engine instead of at the nose of the airplane between the propeller and engine where previous types have been located.

In the original supercharger the turbine wheel was located between the compressor and the nozzle box, which gave a somewhat restricted passage for the exhaust gases, but in the new arrangement the nozzle box is located on the outside of the centrifugal compressor, thereby permitting the operation of the turbine wheel in the outside atmosphere and the discharge of the exhaust gases into the stream of air back of the propeller. This carries the exhaust gases out of the way of the aviator, eliminates gas straying into the cockpit, and gives improved cooling for the turbine wheel.

In all airplane work it is exceedingly important to decrease the weight of all of the apparatus, and this has been borne in mind in the various stages of supercharger design. The original nose-type

supercharger, developed during the war, added 150 lb. to the weight of the plane, whereas the side-type supercharger adds only 100 lb., a great many of the parts being made of aluminum and duralumin.

A geared supercharger for 7500 ft. altitude was also developed, and, due to the small size and refined design and to the nature of the material used in the impeller, it was possible to develop a drive to withstand the severe acceleration conditions involved.

The metal magnesium, which has two-thirds the density of aluminum, was used for the construction of the supercharger compressors, thereby effecting a considerable reduction in weight as compared with even the lightest suitable materials previously utilized for this purpose. Magnesium is beginning to occupy a definite place as an engineering material, as it can now be obtained in commercial quantities. (*General Electric Review*, vol. 28, no. 1, Jan., 1925. Article under the title, Some Developments in the Electrical Industry during 1924, by John Liston, Publicity Dept., General Electric Co., pp. 4–58, 103 figs., the part referring to the supercharger being on pp. 6–7 and illustrated by figs. 4–5, *ad*)

AIR MACHINERY

The Development of the Turbo-Compressor

WITH the adoption of higher rotational speeds and reduction in the number of impellers the internal surface of the compressor has been decreased, which has made adequate cooling by means of a water jacket more and more difficult. Theoretically the various methods of cooling are all equally effective, but the use of intercoolers is becoming more and more general.

In this connection a very common mistake is pointed out which consists in employing an approximate formula assuming that the compression is polytropic and is expressed by an equation of the form pv^n = constant. From this it follows that a compression curve, such as 3 (Fig. 1, bottom), is assumed corresponding to the temperature curve 3 (same figure, top). As a matter of fact, however, when a water jacket is employed for cooling it does not begin to take effect until the temperature of the air reaches a certain value. The temperature then varies as shown in curve 4 (top), the corresponding compression curve being curve 4 (bottom).

The difference between the curves 3 and 4 compared with that between the curve 3 and the isothermal curve 1 is so great that it can be seen at once that the formula based on pv^n = constant is not admissible.

Besides cooling, the design of impellers and diffusers remains one of the principal factors determining the overall efficiency of the machine. It is of advantage to work with the high coefficient of flow, taking, for example, $\delta = 0.1$ as the lowest limit allowable. The difficulty lies in effecting a compromise between the technical ideal of high efficiency and the necessity for keeping the cost of manufacture down to a reasonable figure.

From what the author states, a high value of δ exercises an important influence upon the design of turbo-compressors. An impeller for which δ is large is capable of delivering more air than one for which this value is small; consequently the available inlet area for the former impeller must be greater than for the latter, which means either a shaft of smaller diameter or a larger intake opening. The second alternative results in a reduction of the space available for the blading, which involves a number of difficulties and is impossible to deal with more fully in the present article. It is easily understood, however, that gradually divergent channels reasonable length are more conducive to the delivery of the air under favorable conditions. Moreover, with high-speed single-cylinder compressors, it is found necessary to fit shafts of large diameter, even when they run above the first critical speed, which is the practice of most designers. Here already is the first difficulty to be overcome. Further, the volume of the air decreases in proportion to the compression, so that, if it is desired to maintain a high value of δ, the impeller of each successive stage should be smaller than the preceding one, as may be seen from Formula [1]:

$$\delta = \frac{Q}{u_2 R_2^2} \quad [1]$$

where Q = quantity of free air dealt with, in cubic meters per second

u_2 = peripheral speed of the impeller at outlet, in meters per second

R_2 = outside radius of the impeller in meters.

As the peripheral speed falls off with the diameter, the number of impellers necessary for a given final pressure mounts up rapidly, which naturally results in a greater distance between the bearings, together with a corresponding increase in shaft diameter, or a great number of cylinders. On this account designers have to be satisfied with a value of δ very considerably less than 0.1, and this is particularly the case when the delivery volume is small or the speed low.

To overcome these difficulties higher and higher speeds have been employed—as high as 350 m. per sec. (1148 ft. per sec.). This makes it necessary to employ materials of greater strength and also to take certain special precautionary measures. For instance, Brown, Boveri & Co. subject every impeller for their turbo-compressors to severe overspeed tests during manufacture.

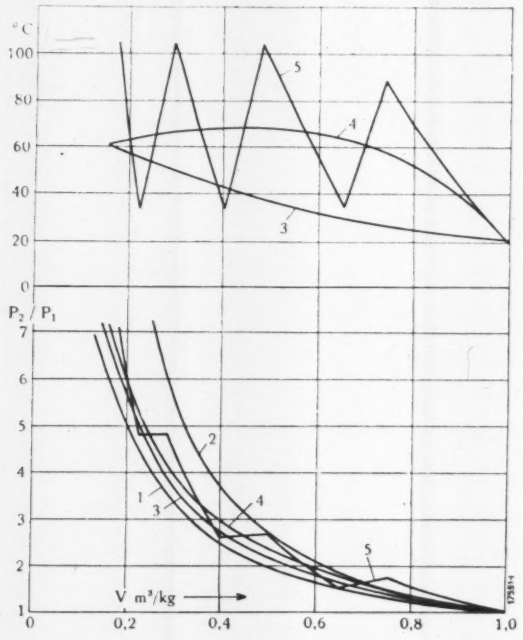


FIG. 1 COMPARATIVE CURVES OF COOLING BY INTERNAL WATER JACKET AND BY EXTERNAL COOLERS

(1, Isothermal compression; 2, adiabatic compression; 3, polytropic compression ($pv^n = C$); 4, actual compression; 5, compression with three external intercoolers. 3, 4 and 5 are the corresponding temperature curves.)

The author proceeds next to a discussion of the mutual influence of the rotational speed, the diameter and bore of the impellers, the efficiency, and the dimensions of the centrifugal machines. In the course of this investigation an interesting table is obtained to show the relation between several of these magnitudes.

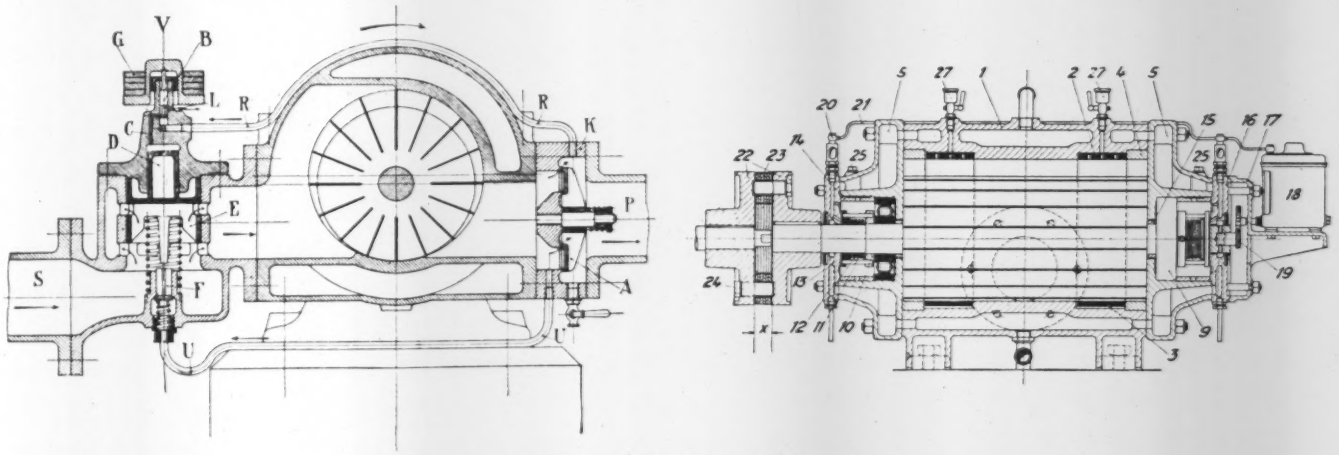
Among other factors affecting compressor design are existing market conditions and steam-turbine developments. As regards the latter, it is stated that Brown, Boveri & Co. are developing the steam turbine along such lines that they find it necessary to divide the machines into two or three cylinders (compare MECHANICAL ENGINEERING, vol. 46, p. 40), and it is quite natural for the compressor design to progress in the same direction, which means a return to low speeds for small delivery volumes. In another direction it is necessary to combine the advantages of high speeds and a large number of impellers. Because of this the new Brown-Boveri compressors designed for dealing with small volumes of air are of the two-cylinder type, while the single-cylinder design is retained for larger outputs. For these impellers a new special type of impellers were developed, and curves in the original article show their efficiencies which appear to be quite high.

As regards the question of regulation of turbo-compressors, the Brown-Boveri concern still advocates regulation by movable diffuser vanes. The question of working on loads below the critical value (i.e., the peak of the pressure-discharge curve), has led to evolving impellers specially suited to operation over an extensive range of regulation. These impellers are not described in the original

article, but curves are given of pressure and efficiency plotted against output obtained with the impellers. (*The Brown, Boveri Review*, vol. 12, no. 1, Jan., 1925, pp. 3–8, 4 figs., *tg*)

Tests of a Swiss Rotary 500-hp. Compressor

DATA of tests carried out on a compressor built by the Swiss Company for the Construction of Locomotives and Engines at Winterthur. The design of the compressor is shown in Figs. 2 and 3. The rotor is located eccentrically and is provided with a number of sheet-steel blades easily moving in slots provided for this purpose in the rotor and bearing at all positions on the walls of the chamber. The centrifugal force on the blades is counterbalanced by two rings rotating with them. These freely moving rings are given a radial play externally, forming a free annular space divided by the blades into cells or small chambers connected with the working cells by ports cut in the rings and serving to equalize the pressure. A hit-or-miss governing device (Fig. 2) comes automatically into action as soon as the desired pressure is exceeded. When this happens the compressed air flowing through the upper conduit lifts the piston



FIGS. 2 AND 3 SWISS 500-HP. ROTARY AIR COMPRESSOR
(1, Cylinder; 2, movable rings; 3, sliding blades in the movable rings; 9, roller bearings; 18, oil pump; 22, 23 and 24, flexible coupling.)

B which is ordinarily held down by the load *G*. This uncovers the passage *C* and the compressed air lowers the piston *B* which permits the slide valve *E* to close the intake passage. Simultaneously with this the poppet valve *F* opens to give passage to the compressed air which is still in the machine and this air escapes through *U* into the intake pipe. Because of the difference of pressure, valve *P* closes and the compressor runs at no load until the pressure in the exhaust pipe *P* [It does not appear clear why both the valve and the exhaust pipe are denoted by the same letter.—EDITOR] comes down sufficiently to permit the piston *B* to descend and connect passage *C* with the exterior air through port *L*. When this is done the compressed air, such as may be on top of piston *D*, can escape. The slide valve *E* opens under the action of the spring, the valve *F* closes, and the operation starts over again.

A single-stage compressor may be used for outlet pressures up to 4 atmos. but for higher pressures a two-stage compressor has to be employed. The air intake cooler between the stages may be located on the same foundation as the machine itself.

The company has recently built a number of large units and has delivered to foreign coal mines compressors of 500 hp. The author, a professor in the Swiss Federal Polytechnic School, had occasion to carry out in June, 1924, acceptance tests on one of these compressors, the results of which are given in Table 1. From these it would appear that, everything considered, the isothermal efficiencies are satisfactory and are comparable with those of reciprocating compressors, assuming that the latter are tested in the same manner as the rotary compressors were, which means by directly measuring the input by a dynamometer and the output by proper apparatus. Actually, however, reciprocating compressors are judged by their indicator diagrams, which the author does not consider sufficiently precise for this purpose. The rotary compressor has, however, certain special advantages in addition to efficiency, i.e., it requires only light foundations, eliminates belting, and is cheap to install, operate and maintain. (Prof. C. Ostertag in *Bulletin Technique de la Suisse Romande*, vol. 51, no. 2, Jan. 17, 1925, pp. 14–16, 3 figs., *de*)

TABLE 1 DATA OF TESTS ON A SWISS 500-HP. ROTARY COMPRESSOR

Test No	1	2	3
Air pressure at the outlet flange of the 2nd stage, atmos	7.0	6.0	5.0
Air pressure at the outlet flange, 1st stage, atmos	1.98	1.88	1.80
Pressure below atmospheric at the intake flange of the 1st stage, mm. of water	308	315	317
Ratio of pressures (preliminary pressure, 723 mm.)	8.38	7.34	6.29
Volume of air taken in, cu. meters per hr	4041	4100	4100
Volume of air taken in, cu. ft. per hr	142,640	144,730	144,730
R.p.m.	422	422	422
Power at the shaft, metric hp	489	454	415
Power at the shaft, American hp	482	448	409
Power, isothermal, metric hp	302	289	267
Power, isothermal, American hp	288	285	263
Isothermal efficiency[1] (see footnote), per cent	61.8	63.6	64.4
Volumetric efficiency of 1st stage, per cent	92.3	93.7	93.7
Rise of air temperature, in deg. cent.	106	97	84
Rise of air temperature, in deg. fahr.	194.4	174.6	151
Rise of water temperature, in deg. cent.	8.5	8.0	7.5
Rise of water temperature, in deg. fahr	15.3	14.4	13.5
Volume of cooling water, cu. meters per hr.	22.3	22.3	22.3
Volume of cooling water, gal. per hr.	5891.7	5891.7	5891.7
Heat Balance (in metric hp.; 1 metric hp. = 0.9863 American hp.):			
In the air	180	167	146
In the water	300	284	265
Losses (conduction, radiation)	9	3	4
Total	489	454	415

[1] The isothermal efficiency is the ratio between the energy necessary to compress to the final pressure the volume of air taken in without any increase of its temperature and the energy actually absorbed by the compressor and measured on its shaft.

FORGING

Upsetting Machine in the Fabrication of Forgings

A PROCESS adaptable for certain classes of small forgings and particularly advantageous where shafts have to be provided with

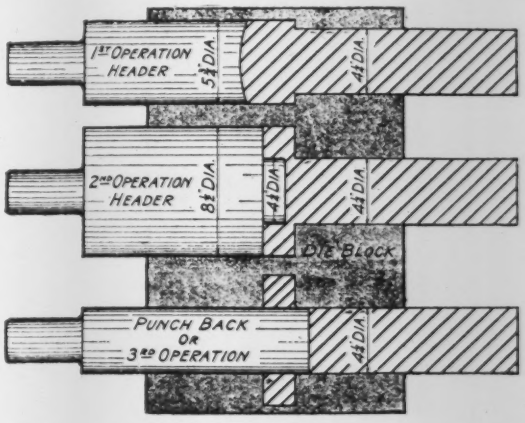


FIG. 4 OPERATIONS IN PRODUCING A RING FORGING IN THE UPSETTING MACHINE

flanges or enlarged sections at one end, but available for use for many other parts. Thus rings for ball or roller bearings as well as gear blanks are frequently made in upsetting machines.

The method by which simple ring forgings are made is illustrated in Fig. 4, which is taken from a recent publication of the American Forge Company, Chicago. To produce a ring of $8^1/_2$ in. outside diameter, $4^1/_4$ in., inside diameter, and $2^1/_4$ in. face, round bar stock of $4^1/_4$ in. diameter is used.

The dies are in three units, viz., a stationary die in which the operator places the stock; a movable die, perfectly matched, which moves forward a distance equal to the length of the stock required to make one forging, and also grips the stock to prevent slipping; and the heading tools that force the stock into the different die impressions.

Referring to the illustration, in the first operation the stock is upset to $5^3/_4$ in. diameter; in the second operation it is upset to the full $8^1/_2$ in. diameter and the forging pierced part way, and in the third operation the forging is pierced completely and the bar pushed free from it, after which the latter is returned to the furnace for reheating. One of the advantages of upset forgings is that they require no draft and that, as a consequence, they usually can be made more closely to size than other kinds.

Among other things which can be made by the same process are crankshafts with integral flywheel flanges, and rear-axle shafts for cars with the spur-type differential gear which have the so-called side gears of the differential forged integral by upsetting. A part that seems to lend itself particularly well to production in the upsetting machine is the transmission drive shaft with the pinion of the constant-mesh set of gears. The pinion is flanged hollow, because it has to be bored out for the pilot bearing. At the plant of the American Forge Company in Chicago such transmission drive shafts are produced on a 7-in. upsetting machine at the rate of 650 per day. (*Automotive Industries*, vol. 52, no. 6, Feb. 5, 1925, pp. 226–227, 4 figs., *d*)

FUELS AND FIRING (See also Internal-Combustion Engineering: Cooperative Fuel Research; Natural Gas as Motor Fuel)

Influence of Pulsations on Combustion

IF A PARTICLE of carbon be suspended in still air and gravity has no effect on it, and further if the conditions of temperature and pressure are suitable, the combustion will proceed in an imperfect manner. It will begin upon the whole surface, but the carbon will be immediately isolated from the air by a layer of

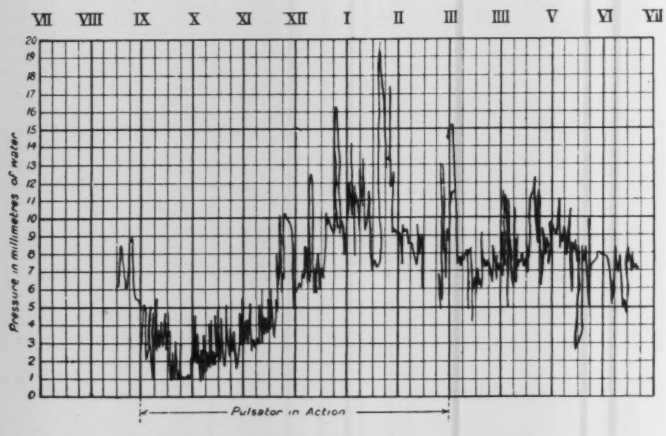


FIG. 5 RECORD OF PRESSURE IN GAS MAIN

cinders and by a layer of burning gases from which it cannot escape because gravity is assumed to have no effect and the air is perfectly still. The combustion will therefore stop. The same effect will arise if a particle of carbon is carried in a current of air at precisely the same velocity as the air itself, no matter what the velocity may be. In order that combustion may continue, it is necessary that there shall be relative movement between the air and the carbon. In particles this relative movement is effected by the obstacles in the path of the burning gases, by changes of volume and of pressure in the gas at different points, and by difference of pressure or velocity produced either naturally or artificially by pulsations. The questions which the author considers are the various means to produce these lateral movements of the air between the particles of the coal.

Among other things he points out that if the relative speed of the air and the particles of carbon is continually changing, the film of burned gases charged with distillates is carried away and new air reaches the particle of carbon. It may be observed that the speed of the particle of carbon will change with that of the air, but that is true only in part as the inertia of the particle must be considered, and moreover the act of combustion sets up lateral movements of the particle on its own account. It is for that reason that temperature has such an important influence on the activity of combustion. These rapid pulsations play their part by modifying continuously the conditions of combustion, and by attacking the dynamic equilibrium which, as a consequence, is also able to persist for relatively short periods. In some remarkable experiments on the combustion of powdered fuel in suspension, carried out by Messrs. Taffanel and Audibert, certain tests were made in a tube in which the speed of combustion increased regularly up to the point where an undulatory movement of a frequency compatible with the characteristics of the tube was reached. At that moment the speed of combustion

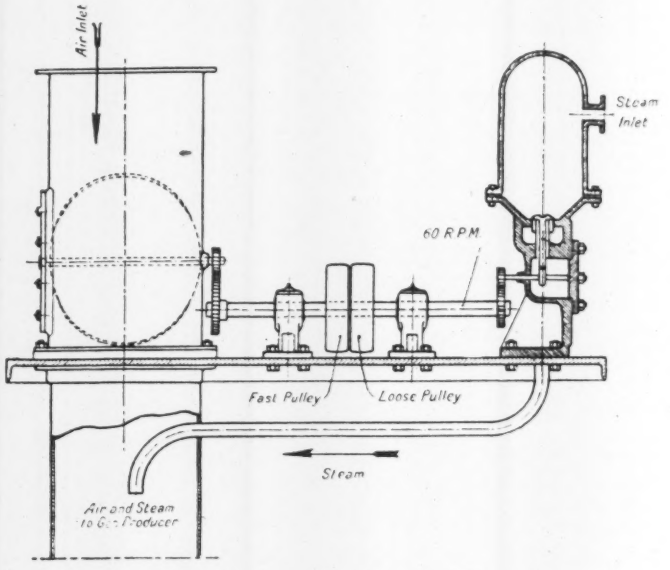


FIG. 6 PULSATOR FOR USE WITH GAS PRODUCERS

increased much more rapidly and irregularly. Indeed, the combustion became explosive. In any case, Audibert's experiments at Montlucon have shown variations in the length of the flame between 15 and 50 per cent as the pulsations varied between 700 and 2100 per min., but these figures must be regarded with some reserve, for, so far, we have nothing to indicate that 50 per cent is the maximum, and it may certainly be said that the frequency of the pulsations must depend upon the speed of the air charged with carbon and upon other characteristics.

The author made some experiments on a gas producer of the Bouchayer-Hilger type, 2.5 m. in diameter at the grate level. Fifty pulsations a minute were given, the pulsations being divided in a rhythm of four. During the first two air alone was blown in, during the third air and steam were introduced together, and the pressure on the grate—which in the two proceeding phases was that due to the draft alone—increased owing to the pressure of the steam. During the fourth phase air and steam were stopped altogether, and the pressure on the grate fell rapidly until it became equal to that in the delivery main. It must be understood that these differences of pressure are, as a matter of fact, very small, not more than 1 per cent of the prevailing pressure. Nevertheless the result was absolutely conclusive and fulfilled the author's prevision. Fig. 5 gives an autographic record of the pressure in the gas main at a place about 50 m. from the producer. Before the pulsations were applied the pressure oscillated between 5 mm. and 10 mm. As soon as the pulsator was started the pressure fell considerably, because the zone of combustion was not modified, while the air was stopped once in four, and that of the steam three times out of four. But it must be observed that, with the same blowing pressure, the

pressure rose slowly and regularly until it far surpassed the preceding pressures. Quite a considerable time passed before this effect became noticeable—about three hours with air and steam and six hours with air alone. Successive tests all gave similar results. Comparative tests which have been made at Commentry with coal having a high percentage of ash, alternatively with and without pulsations, have shown a considerable economization of coal. The consumption of steam is also considerably reduced because the steam jets only open once in four. Poking is also reduced, and, as a matter of fact, it was impossible to employ coal of this kind without pulsations. Fig. 6 shows the type of pulsator employed for gas producers, and requires no explanation. (J. Deschamps in *The Engineer*, vol. 139, no. 3606, Feb. 6, 1925, pp. 152–153, 2 figs., *te*)

GAS PRODUCERS

The Gerard Gas Producer

ONE of the essential features of the producer designed by the French engineer, Céleste Gérard, is that the grate is formed of three plates superimposed one over another. The distribution of the air is effected through the joints between these plates and can be

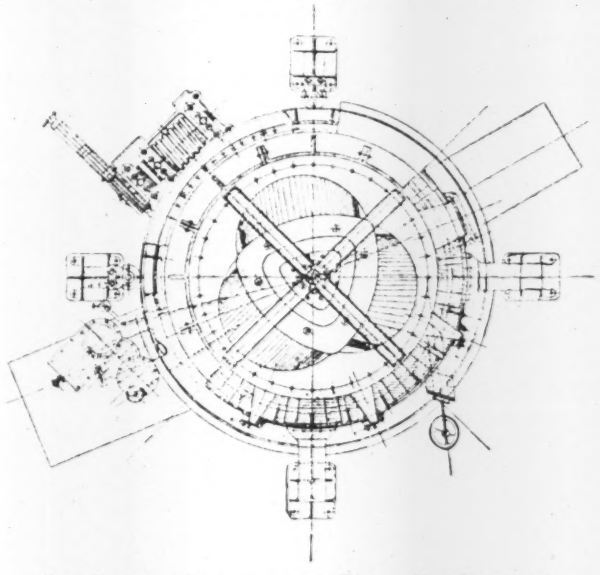


FIG. 7 CROSS-SECTION OF THE GÉRARD GAS PRODUCER

judiciously spread over the entire surface of the producer. The rotating body which supports the grate is formed of a triangular prism having curved faces each of which is provided with a semicircular inclined plane (see Fig. 7). This special form has two functions. First, the inclined plates continually lift the mass consisting of ashes and combustible matter and then let it drop suddenly and this continuous stirring improves the contact of the air with all the particles of the combustible matter and impedes the formation of clinker. Second, the inclined curved faces of the prism are equipped with pointed projections, and, to all practical purposes, form the movable part of a crusher, of which the fixed part is the lower surface of the hearth of the producers formed of heavy cast-steel elements likewise bristling with projecting points. The crusher permits only fine cinders to escape through it and this results in such a complete combustion of all combustible material in the ash, that the latter when coming out of the producer contains not more than 0.1 to 0.2 per cent of carbon.

The mechanical spreader consists of a water-cooled crosshead which moves in a horizontal plane at a suitable distance from the grate. The motion of the crosshead is produced through its connection with the rotatable part of the grate. The steam necessary to operate the producer is generated through the heating of the cooling water of the crosshead above referred to. This water partly transformed into steam reaches into the rotatable part of the grate and then humidifies the air of injection. It is stated that excellent results have been obtained with the Gérard apparatus on all kinds of fuel. (*Chaleur et Industrie*, vol. 5, no. 56, Dec., 1924, p. 654, 2 figs., *d*)

INTERNAL-COMBUSTION ENGINEERING

Natural Gas under Very High Pressure as a Motor-Vehicle Fuel

PARTICULARS of work done in France by the company operating what appears to be the only important source of natural gas in the country, namely, that at Vaux-en-Bugey. The gas as used for motor-vehicle fuel is furnished in steel bottles at a pressure of 150 kg. per sq. cm. (2100 lb. per sq. in.), and is expanded in a regulator valve, from which it escapes at a uniform pressure of 15 mm. (say, 2 in.) of water. Thence it passes through a specially designed gas carburetor where it is mixed under the action of the motor suction with the necessary amount of air. From this gas carburetor the mixture goes into a gasoline carburetor, which, however, does not contain any gasoline. When the motor is operating on gasoline the gas bottles are closed and the gasoline carburetor connects with the gasoline tank. The method of mixture governing described in general in the original article is such that the process is entirely automatic and the driver may not know whether he is operating on gasoline or gas. The device has been installed on several vehicles, among others, a Berliet four-ton truck and a Nash Quad tractor. It is stated that the truck ran on a single charge of gas (four steel bottles) 125 km. (78 miles). The bottles can be charged with as much ease as a gasoline tank or may be replaced by filled bottles.

As regards the comparison between gas and gasoline, it is stated that 1 cu. m. of gas (35.3 cu. ft.) = 1.20 liters (3.1 gal.) of gasoline of a light type, and it is claimed that the price of the two fuels being equal, there is an economy of 50 per cent from the use of gas as compared with gasoline. It is stated, however, that gas can be economically used only in localities where it is naturally available, as the transportation of the bottles to great distances is excessively costly. A. Godel in *Chaleur et Industrie*, vol. 5, no. 56, Dec., 1924, pp. 614–618, 7 figs., *g*)

Recent Cooperative Fuel-Research Progress

GENERAL report on the work done at the Bureau of Standards in coöperation with and at the commission of the American Petroleum Institute, The Society of Automotive Engineers, and the National Automobile Chamber of Commerce. The direction of the work is indicated by a resolution of the Steering Committee in respect to the investigation from July, 1924, to July, 1925, which recommended (1) that the factors contributing to crankcase-oil contamination be investigated; (2) that the factors contributing to easy starting be investigated; and (3) that an investigation of vaporization be made to coördinate design of internal-combustion engines and the volatility of the fuel.

The starting point of an investigation of the subject of vaporization consists of obtaining more exact knowledge of what constitutes effective fuel volatility in both the temperature range covering crankcase-oil dilution and in the initial range which determines starting the engine. The conclusion to which the experimenters have arrived so far is that under average operating conditions crankcase-oil dilution (1) depends primarily on the average temperature of the cylinder walls; (2) is likely to reach an equilibrium value if starting periods are not too frequent; (3) depends directly on fuel volatility; (4) is directly dependent upon the average fuel-air ratio; (5) does not depend much upon the piston temperature; and (6) is not dependent much upon the charge temperature or the degree of vaporization.

As regards the remedies thus far, the "don'ts" are mainly emphasized, namely, increasing the maximum temperature is likely to involve increased trouble from detonation. Both high jacket temperatures and high oil temperatures mean that the oil must be of rather high initial viscosity in order that its viscosity shall be adequate at its normal operating temperature. Any increase in oil viscosity means increased difficulty in cranking the engine when it is cold, and greater friction losses before the engine reaches its normal temperature. There do not seem to be any objections, however, to a decrease in the time required for the engine to reach its normal temperature, and such a decrease is greatly to be desired. (S. W. Sparrow, Mechanical Engineer Bureau of Standards, Mem. A.S.M.E., and J. O. Eisinger, Asst. Mechanical Engr., Bureau of Standards. *The Journal of the Society of Automotive Engineers*, vol. 16, no. 2, Feb., 1925, pp. 237–242, 11 figs., *e*)

The Missing Pressure in Gas Engines

A PAPER containing an account of experiments made with the object of obtaining a quantitative solution of the problem of the missing pressure in gas engines. The author starts with a historical survey of the problem. A study of previous work led him to believe that the solution of the problem of the missing pressure was practically impossible by means of experiments of a more or less direct nature and he accordingly adopted an indirect method of experimenting which is that of determining the internal energy of a weak mixture of coal gas and air exploded in a closed vessel in the neighborhood of its maximum temperature and comparing it with that of a strong mixture when it had cooled to the same temperature as the weak mixture had at that time.

It will be recalled that the problem of missing pressure in gas engines goes back to the earliest experimenting on internal-combustion engines, when it was noticed that the pressures developed on on explosion of the charge were less than half those which were calculated from the known heating value of the fuel on the assumption that the working fluid (which contains normally more than 80 per cent of nitrogen) behaves like ideal air.

Among other things, it was found in the course of the author's experiments that combustion is practically complete about a quarter of a second after the moment of maximum pressure, but the rate of pressure during this interval is not constant and becomes less and less as cooling proceeds. Moreover about half of the 10 per cent of coal gas remaining unburned at the moment of maximum pressure appears to be burned within $^1/_{20}$ of a second.

The author's experiments show that even under the ideal conditions obtainable in a closed vessel some 10 per cent of the gaseous fuel (coal gas) is unburnt at the moment of maximum pressure, and that, therefore, incomplete combustion, in addition to increasing specific heat and heat loss, must be regarded as an important factor in limiting the pressures developed.

An examination of the gas-engine experiments of Clerk and Hopkinson in the light of the closed-vessel experiments shows pretty clearly that incomplete combustion is of even greater importance in limiting the pressures developed in an engine cylinder. About 30 per cent of the gaseous fuel appears to be unburnt at the moment indicated by the peak of the indicator diagram when the ignition setting is normal. The explanation of this large amount of unburnt gas at this moment would appear to be that in general the peak of the diagram is reached when the flame has just filled the combustion space, at which moment closed-vessel experiments indicate that about 75 per cent of the gaseous fuel has been burnt. During the later stages of inflammation the rate of rise of pressure in a closed vessel is at its maximum value, but during after-burning (which has been defined as the burning which takes place after complete inflammation) the rate of pressure slows down, and it is reasonable to suppose that the rise of pressure due to after-burning which would have taken place had the piston remained stationary at its inner dead center is masked by the expansion due to the motion which actually occurs.

The experiments described, while they emphasize the importance of incomplete combustion equally with that of increasing specific heat as a factor in limiting the pressures developed, do not indicate that it very greatly affects the thermal efficiency of gas engines working upon strong and moderately strong mixtures, for the bulk of the after-burning is completed in the early stages of the expansion stroke. This being so, after-burning is an advantage because it reduces the ratio of the maximum pressure to the mean effective pressure—a factor having a special significance in mechanical design.

These experiments do not therefore call for appreciable amendment of figures given by various experimenters for the efficiency ratio (that is, the ratio of the actual indicated thermal efficiency to the ideal possible), which may approach to 90 per cent in well-designed engines. Little margin is therefore left for improvement as long as the standard cycle of operation is adhered to. (W. T. David, Professor, University of Leeds, in *Proceedings of the Institution of Mechanical Engineers*, no. 5, 1924, pp. 741–762, 8 figs., *teA*)

The Andreau Motor

ONE of the disadvantages in the cycle of operation of the ordinary internal-combustion engine, in particular, the four-stroke cycle, is that the length of travel of the piston in each stroke is equal to that in every other stroke, but is not in strict correspondence with the requirements of the cycle. Thus, for example, during the expansion stroke the elasticity of the gas is but poorly made use of. The exhaust begins as a matter of fact when the pressure has dropped to about 3 kg. (42.6 lb. per sq. in.) and the gases have still a temperature in excess of 700 deg. cent. (1292 deg. fahr.). Because of this not only is a part of the energy of combustion entirely lost but certain undesirable effects are produced such as excessive heating of walls, valves, etc. and an excessively noisy exhaust.

The essential feature of the Andreau motor is that it has completely set aside this method of operation and has substituted for it a simple mechanical combination which permits making the length of the piston stroke different in different parts of the cycle and, in particular, makes the exhaust stroke longer than the intake stroke. This, in its turn, insures a perfect withdrawal of exhaust gases from the explosion chamber and a longer duration of expansion than of compression, and this in its turn produces a material increase in the work delivered by the motor.

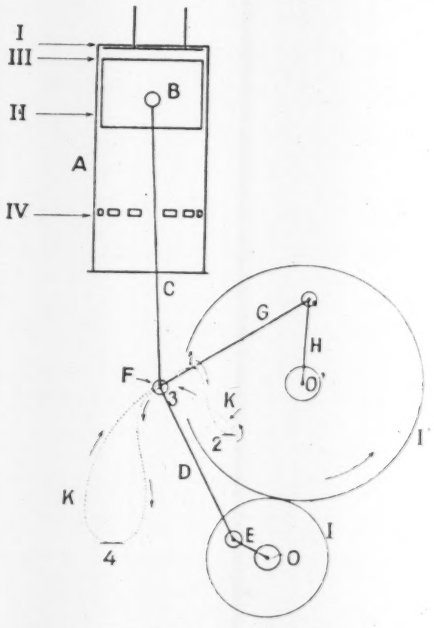


FIG. 8 GENERAL MECHANICAL SCHEME OF THE ANDREAU MOTOR

The basic features of the scheme of operation are shown in Fig. 8. The piston B moves in the cylinder A, while the connecting rod C connects at F to the tip of an articulated system which comprises a small connecting rod D articulated at F at one end, and at the other end, operating crank E working with the motor shaft O.

At the same point F, however, a second connecting rod G is articulated which at its other end is attached by a pin to a crank H rigidly connected on to a second shaft O' parallel to the shaft O. The shafts O and O' are rigidly connected to each other by a system of gearing which imposes on the shaft O' a speed one-half that of O. On considering the motion of the point of articulation F produces when the two shafts are running and one is driving the other, it will be found that this point F describes a curve K which seems to be quite complicated and which obviously depends on the dimensions of the various elements constituting the articulated system. If these dimensions have been properly selected the curve will be the shape shown in fine dotted lines in Fig. 8.

While the shaft O makes the two revolutions which correspond to the four strokes of the cycle, the point F describes the curve K in the direction of the arrows, and the points 1, 2, 3, and 4 where the tangents to the curve are horizontal, correspond to the dead centers. When the articulation point F is at 1, the piston B occupies the position I at the very top of the cylinder; while F describes the arc from 1 to 2 the piston descends from position I to II, etc. the Arabic numbers on the curve corresponding to the respective Roman numbers indicating the piston positions. A brief glance at the figure will indicate, however, that the strokes are unequal.

It may be of interest now to consider what happens in the cylinder during this cycle. To do this it will be necessary to consider Fig. 9 jointly with Fig. 8.

Intake. At first the piston occupies the position I at the top of the cylinder, which is built without a valve box. Because of this design fresh gases when they enter the cylinder do not come into contact with the products of combustion, the latter being completely expelled during the preceding stroke. As a further result of this condition of operation a complete filling of the cylinder is effected at a low temperature; to do this a large intake valve has to be pro-

vided so as not to oppose the flow of gas at the beginning of the stroke. The piston then moves down to position II, which is the end of the intake stroke and which determines the amount of mixture taken in. It is claimed that the admission is more complete than in an ordinary motor of the same bore and power running at the same speed and having a stroke twice that of the Andreau motor.

Compression. The piston rises from position II to position III and compresses the mixture to the desired extent. The space between I and III represents the compression chamber. The actual displacement of the piston is here claimed to be less important than in the conventional engine, since on account of the fact that the filling of the cylinder is more perfect, higher compression may be obtained.

Expansion. After the explosion the piston descends from position III to position IV. As the figures show, this stroke is very long as compared with the compression stroke. On the other hand, it will be noticed that during this stroke, which is the only energy-producing stroke of the motor, the point F describes the arc 3–4 of the curve K in Fig. 8, and that in its major part this arc roughly coincides with the axis of the cylinder. In other words, the connecting rod is practically free from obliquity during this part of the cycle and this does away with the greater part of the friction of the piston against the cylinder wall. In the conventional motor, on the other hand, the connecting rod has the same degree of obliquity during each stroke and the lateral pressure of the piston against the cylinder is at its maximum during the working stroke. The Andreau arrangement not only reduces power losses but decreases the tendency of the cylinder to wear out of round. Because of the very long expansion stroke the gases have the chance to expand down to a pressure of only 100 grams per sq. cm. (1.42 lb. per sq. in.) gage. In this way all the energy contained in the gases between the ordinary exhaust pressure and the low exhaust pressure stated above, is saved.

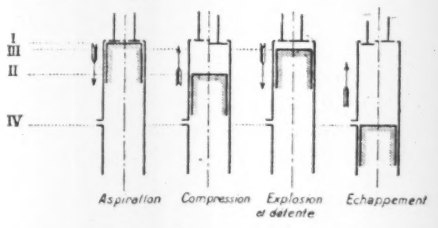


FIG. 9 DIAGRAM SHOWING THE UNEQUAL LENGTH OF STROKES IN THE CYCLE OF THE ANDREAU MOTOR

Exhaust. During this stroke the piston rises from position IV to position I. The difference in the length of the expansion stroke and intake stroke makes it possible to provide a large exhaust port in the cylinder itself at the end of the stroke (at position IV, see Figs. 8 and 9), which permits about 80 per cent of the gases to escape through that port. In moving upward the piston closes this exhaust port and forces such gases as may remain in the cylinder out through the exhaust valve on top of the cylinder.

Tests have been made in the Laboratory of Arts and Manufactures in Paris on a single-cylinder Andreau motor built with funds provided by the National Office of Research and Invention. This motor after having been operated for 1500 hr. showed no appreciable wear, this applying particularly to the triple articulation F of Fig. 8. The tests have indicated that in an engine of automobile or truck size a fuel consumption of 160 grams of ordinary gasoline (0.35 lb.) per hp. may be confidently expected, and that this may be lowered for aviation motors to from 110 to 120 gr. (0.24 to 0.26 lb.) per hp. This would be essentially the consumption of a semi-Diesel motor enjoying the advantages of running on gasoline fuel. A general consideration of the design would also indicate that there are no especially difficult shop problems involved in the production of this motor. The inventor has received for this motor the first Jean Barès prize of the National Office of Research and Invention. (E. H. Weiss in *La Nature*, no. 2651, Jan. 24, 1925, pp. 55–59, *dA*)

MACHINE SHOP

The Scope of the Linisher

IT IS claimed that the use of the linisher as an aid to production is not fully appreciated, but that it can, if correctly equipped and applied, perform better and cheaper work which could not be done as well on another machine, and, in addition, will produce a grained finish with a flat surface and square corners.

The author cites several cases of parts finished on the linisher (Fig. 10), for example, the two-pin plug used extensively in connection with wireless apparatus shown in the upper left-hand corner of the figure. At first the part was made from ebonite purchased to the same section as the plug body, the bodies being cut off to length on a milling machine, drilled in a jig, and the plugs inserted. Due to the variation in the size of the raw material the parts would not fit the drilled jigs correctly. Moreover, in linishing the outsides of the plug bodies to obtain a matt finish, a high percentage were scrapped through their not being of symmetrical form. Next, the use of sheet ebonite was tried, the material being cut into strips and the edges milled to the required radius. This made the section of material correct to size, but did not eliminate the large finishing rejections. Molding the parts did not solve the problem either, but the simple tool equipment used on the linisher shown in the center view of Fig. 10 successfully solved the problem. The mandrel is in two parts, A and B, which have respectively holes and plugs to suit the two-pin plug to be linished. The section of the

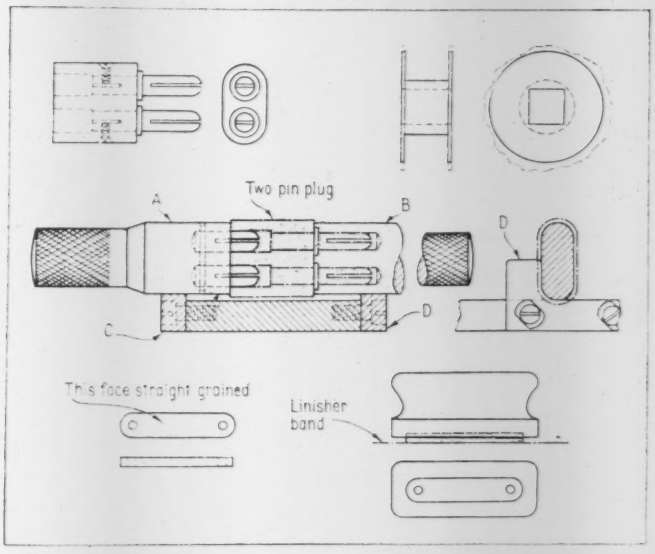


FIG. 10 TYPICAL PARTS FINISHED ON THE LINISHER AND SIMPLE LINISHER TOOL EQUIPMENT

mandrel is symmetrical about the same centers and center lines as the plug body, but both parts are made 0.125 in. smaller all around.

This two-part mandrel is attached to the plug body so as to form one continuous mandrel, and it will be realized that when this mandrel is rotated on the linishing band until it rests upon the side guides C and D, the plug body will be linished to a form that is the same as the section of the mandrel, except, of course, that it will be larger, the exact size being controlled by the height that the guides C and D are set above the band. The vertical adjustment to these side guides is effected by the inclined slots as shown in the side view. Moreover, these side guides are provided with an abutment for the mandrel, to take the thrust of the linishing.

This equipment proved so successful that the same principle was employed to linish other molded wireless components. Due to the previously mentioned defects, the bobbin shown in the top right-hand view in Fig. 10, when molded, had a slight radial flash on the edge of the cheeks, as shown by the broken lines. This was effectually removed by the same method.

The lower views in Fig. 10 show a switch lever also successfully manufactured in the linisher, and another figure in the original article shows a linishing fixture for holding and working parts. (N. G. N. in *Engineering Production*, vol. 8, no. 149, Feb., 1925, pp. 34–35, 2 figs., *p*)

MACHINE TOOLS (See also Machine Shop: Linisher Work)

A Gear-Cutting Hob of Novel Construction

DESCRIPTION of a hob recently placed on the market by the O. K. Tool Holder Company of Shelton, Conn. It is made up of five

sections, each section being a single drop forging comprising one turn of the helix, and each unit being identical and interchangeable with every other unit. Five of the sections are assembled on an arbor with radial keys in place, and are drawn together with four binding bolts into what is to all intents and purposes a single-piece worm. The unit is then set up in a lathe and the top, bottom, and both sides of the helix are dressed in the same manner as a worm would be cut from solid stock, except that the work of stock removal is avoided.

The teeth in the hob are O. K. tool bits, die-forged to the shape required and held by a locking wedge. The tool bits are all ground to gage and are interchangeable, though not all of them are of the same shape above the pocket. The methods employed in the manufacture of these hobs are extensively described in the original article. (*American Machinist*, vol. 62, no. 5, Jan. 29, 1925, pp. 207–209, *d*)

New Machine for Punching and Cropping Angle Bars for Structural Steelwork

DESCRIPTION of a machine made by James Bennie & Sons, Ltd., Glasgow, Scotland, for the piercing and cropping of large numbers of duplicate light angle bars for structural steelwork. It also serves to facilitate the erection of members by stamping each batch with its identification number or mark.

The machine consists actually of three machines mounted on a common bed, one of them being fixed while the other two are independently adjustable along the bed. The machine on the left-hand end carries an angle cropper to crop the bars to length, a punch slide to which the punches are fitted, and a hammer to stamp figures or other marks of identification upon the bar. The two right-hand machines each carry a punch slide and an end stop, the end stop of the middle machine being used only when short bars with holes in the ends only are required. The movable machines are adjusted along the bed by the two large handwheels at the rear, through worm gear at the right-hand side of rack machine, to rack and pinion in the bed. All three machines are driven from the one common top shaft and all are operated by the one treadle at the front. This treadle may be constantly depressed, in which case the machines operate at 60 strokes per minute; or, if desired, it may be operated so that all three of the machines make single strokes and come to rest with the punch slides in the top position.

The punches are carried in sockets fitted in the slots in the punch slide and are capable of several inches of longitudinal adjustment. The sockets are also slotted transversely so that about an inch of transverse adjustment can be obtained to enable holes to be punched at various distances from the heel of the angle bar.

The dies are arranged to take the angle-bar heel uppermost with the wings each at the angle of 45 deg. to the vertical, the punch slides operating in a diagonal direction at right angles to the wing being pierced.

The bars should be straightened before being put into the machine, but a special arrangement is provided to press the bar down upon the dies before punching, thereby insuring that the holes are always punched at the required distance from the heel even though the bars may not be perfectly straight. This device is in the form of a lever disposed at the left-hand side of each movable machine, and is operated from a cam on the main shaft acting through the intervention of a spring.

As the machine when continuously running makes 60 strokes per minute, the work of punching, cropping, and stamping is performed in one second, and as the entry and removal of the bars occupies only a few seconds (dependent chiefly upon the length and weight of the bars), it is evident that an enormous production of finished bars is easily obtained at a trifling cost and that all bars are exact duplicates of each other.

The original article shows a machine capable of punching two holes $^7/_8$ in. in diameter through angle bars $^1/_4$ in. thick. The angle cropper cuts angles up to $2^1/_2 \times 2^1/_2 \times {}^1/_4$ in. The length of the bed is about 12 ft. which permits of bars being cut to any length not exceeding 10 ft. Flat bars may be punched and cut to length in this machine without any change of tools. (*British Machine Tool Engineering*, vol. 3, no. 31, Jan. & Feb., 1925, pp. 208–209, 1 fig., *d*)

MARINE ENGINEERING

The Corrugated Ship

NOTES from an address by A. H. Haver, inventor of the Monitor corrugated ship, at a meeting of the Wm. Froude Naval Architecture Society in the University Building at Delft, Holland. Four Monitor corrugated vessels have recently been constructed in Holland. A claim is made that a corrugated vessel is not only able to ignore much of the severe handling which would be disastrous to an ordinary ship, but that if the corrugated steel were to be damaged the repair costs would be definitely lower than those of a plain ship damaged under the same conditions. This was explained to be due to two facts: (1) the manner in which damage is localized by the corrugations, and (2) the much smaller amount of shell riveting and absence of the necessity of frame repairs.

The influence of the corrugations upon the performance of a vessel was discussed, but it does not appear that their precise influence has been definitely established. It is claimed that when the engines of a stationary corrugated vessel are reversed the propeller stream can be seen emerging from the fore end of the corrugations in a remarkably stable condition of flow, and it seems probable that an analogous action takes place with the vessel going ahead—and this view is definitely supported by the stable nature of the wake actually observed. This reduced eddy would, of course, contribute to saving of power by a corrugated ship. It is also stated that the bow wave on the corrugated ship has been observed to be markedly flatter than on a similar plain shape; if true, this is equivalent to a reduction of wave-making resistance.

In the discussion which followed, Mr. Vlig, Outside Manager of the New Waterway Shipbuilding Co., Holland, where three corrugated vessels have been constructed, stated that he had found that the corrugated ship offered no constructional difficulties and that while its labor cost per ton of steel was slightly higher than that of a plain ship, yet its cost per ton deadweight was definitely less. Contrary to what might be supposed, it was stated that in the preparations of the corrugations it had been found quite unnecessary to furnace a single plate. (*Shipbuilding and Shipping Record*, vol. 25, no. 4, Jan. 22, 1925, p. 88, *dg*)

POWER-PLANT ENGINEERING

The Ginabat Surface Condenser

THE design of the Ginabat surface condenser was described in an abstract published in MECHANICAL ENGINEERING, vol. 47, no. 2, February 1925, p. 132.

The following are some of the data of tests of Ginabat condensers of commercial size. It may be stated in this connection that a number of French destroyers have adopted these condensers and that there have been already installed on land a capacity of 700,000 kw. and in the French navy of 700,000 hp.

Among the most interesting tests are those carried out at the Central Station of the Electric Energy Company of the North of France at Comines. These tests are interesting because the condensers are absolutely the same in every detail except for the tube arrangement, which is Ginabat in one case and standard in another; both have been working for the same period since their last cleaning; they receive the steam from turbines of the same make driving identical alternators. Under these conditions the best condenser should show a higher vacuum and consequently a lower steam consumption and it would appear that this is exactly what happened in this case. The data of these tests are presented in Table 2.

TABLE 2 RESULTS OF CONDENSER TESTS AT COMINES

(Average figures recorded at same working power)

	Customary arrangement of tubes, unit No. 3	New arrangement of tubes, unit No. 1
Steam weight per hr., kg.	80,080	76,440
Steam pressure at the main pipe, kg. per sq. cm.	18.1	18.0
Superheating temperature, deg. cent.	328	325
Condenser pressure, mm. of mercury	40.85	31
Corresponding temperature, deg. cent.	34.57	29.7
Average temperature of cooling water, inlet, deg. cent.	19.05	19.0
Average temperature of cooling water, outlet, deg. cent.	25.23	24.8
Condenser-water temperature, deg. cent.	29.2	26.16
Vacuum, percentage of the atmospheric pressure, 860 mm., per cent	94.6	95.9
Value of the coefficient *K*, French calories	1336	2110
Value of the coefficient *K*, English B.t.u.	274	433

The tubes of the two condensers were covered inside with a layer of organic matter (refuse from tanyard) which influenced the efficiency. This organic matter is very viscous and leaves deposits upon the screens which are frequently cleaned. The tubes were cleaned eight days before the tests, a solution of hydrochloric acid being used; thereafter the two condensers were working for the same length of time. The duration of each test was one hour, during which time the rate of condensation on the two condensers was constant and absolutely the same, the records being taken every ten minutes. The temperatures were measured by a thermometer graduated to tenths of a degree. The pressures were measured with differential mercury gages.

The characteristics of the two condensers are as follows: cooling surface, 2700 sq. m.; number of circulations, 2; total number of tubes, 7500; dimensions of tubes, 20 × 22 mm. (Rear-Admiral Engr. A. Ginabat in *Mémoires de l'Association Maratime et Aéronautique*, 1924.)

Recent Developments in the Use of High-Pressure and Superheated Steam

THE author presents briefly the theoretical advantages of high pressure and then proceeds to the discussion of the high-pressure apparatus. Certain difficulties have been encountered in the manufacture of high-pressure boilers he states, but these are being gradually overcome through an increased knowledge of the methods of manufacture and materials used.

Recent investigations of boiler materials have shown that heretofore important points were not sufficiently taken into consideration when selecting test bars in the usual manner and making the tests for determining the strength. It was, for instance, found that the strength and composition of plates are poorer at the edges than at the middle. This is an important point in the case of plates for boiler drums, as the riveted seams are made along the weakest parts of the plate, i.e., near the edge. Serious boiler defects have repeatedly occurred due to the failure of plates which at the test seemed to conform with the official regulations, but later proved defective. A careful analysis has also shown that plates which were originally satisfactory underwent alterations in the course of manufacture or under operating conditions. It was found that the pressure exerted when riveting was generally far too high, so that invisible cracks developed in the structure and strength was reduced. The gradual alteration of the strength of a plate which is stretched when cold beyond its elastic limit is also an important factor to be considered. The effect of recrystallization through cold stretching which occurs usually at higher temperatures should not be overlooked in connection with high-pressure boilers. When a boiler is in operation, alterations similar to those mentioned may occur through temperature stresses due to the difference in temperatures at various heating surfaces, or between the inner and outer surfaces of the drum. These damaging influences become more important as the boiler pressure and the thickness of the plates increase, particularly because of the greater difficulties of riveting and machining. Higher evaporation rates have a similar effect, as they offer greater temperature difference, and additional stresses in the material are likely to occur.

The remarkable tendency toward higher pressures which is clearly manifested today will compel manufacturers to find suitable kinds of boiler materials. The capacity of machines and the qualities of materials from which they are to be made are at present the most important questions in modern engineering, and it will probably be some time before these questions are definitely solved, though considerable progress has already been made.

The latest investigations have shown that riveted drums which require shaping and bending of plates when they are cold should not be used for high pressures, say, above 800 lb., and only drums forged from one piece should be applied.

For reasons of cost and design the size and number of drums of high-pressure boilers (say, above 1000 lb.) has to be reduced to a minimum, which, in addition to the small tubes, makes the water and steam space of the boiler insufficient for the ordinary load fluctuations, and it becomes necessary to supplement the boiler capacity by some adequate and inexpensive means of storage such as the Ruths steam accumulator.

As regards superheaters, it is pointed out that with increased pressure and superheat constant temperatures are imperative, as any temperature fluctuations cause additional stresses in the prime mover which may become detrimental. In Europe, special temperature regulators have been designed and are often applied with superheaters for high temperatures. American engineers have approached the problem from another standpoint. Proceeding from the fact that any load variation applies to both boiler and superheater, as the same amount of steam which is generated by the boiler passes to the superheater, the superheat will be constant if the heat absorbed by both boiler and superheater varies in the same proportion. On the other hand, it is a known fact that the first few rows of boiler tubes generate most of the steam, so that a superheater located between or near these tubes will be subjected to the same variation as the part of the boiler where most of the steam is generated, and in this way the ideal conditions for constant temperature are closely approached. This arrangement was first applied in the large 2647-hp. boilers at the River Rouge plant of the Ford Motor Company. The remarkable results obtained with this installation have caused the designer of the superheater to apply the same principle in other types of boilers. Since then a number of installations with superheaters in this location were put in operation, every one of which has borne out this theory.

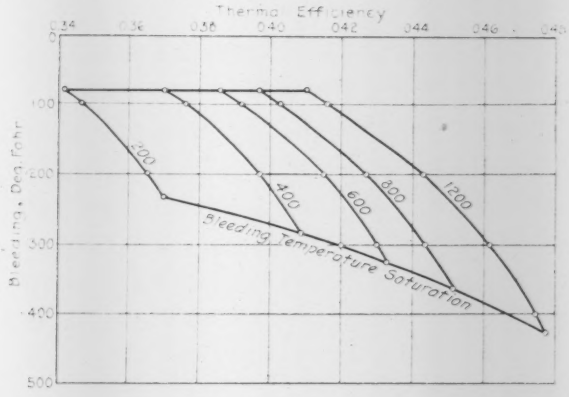


FIG. 11 DIAGRAM SHOWING INCREASE IN THERMAL EFFICIENCY DUE TO STAGE HEATING

With the rapid increase in pressure and superheat the question of steam velocity in the superheater elements has attained greater importance, as the life of the elements and the efficiency of the superheater depend upon the steam velocity in the superheater. The elements of the more recently developed modern superheaters are long enough to furnish high superheat with the high steam velocity. This has been made possible by joining two or more pipes and making them continuous by means of forged return bends which are made out of the metal of the tubes by a special forge-welding process without the use of electric or other autogenous welding.

Headers and joints, such as the ball-clamp joint, have been developed of such a character that when necessary a unit can be removed from the superheater in a very short time and without entering the boiler setting, thereby avoiding the necessity of cooling off the boiler and then firing up again.

From this the author proceeds to a discussion of the design of high-pressure reciprocating engines and turbines, mentioning such types as the high-pressure DeLaval turbine, the Brown-Boveri series turbine, and the Brünn types. He also discusses the subjects of regenerating and reheating. As regards the former, he mentions one advantage which is not generally recognized.

In the matter of turbine design there is an inclination to increase as much as possible the capacity of a turbine for a given number of revolutions. It is limited by the amount of steam that can pass the last blade wheel at a certain vacuum and at certain exit losses. By decreasing this amount of steam, the possible maximum capacity of the turbine is increased. With stage heating, the steam flow through the turbine is gradually decreased, thus decreasing the capacity-limiting factor. This is the more pronounced the higher the initial steam pressure. For instance, at a pressure of about 1500 lb., 47 per cent of the steam must be bled from the turbine in order to heat the water to the maximum temperature.

The efficiency of the regenerative cycle is higher in proportion to the number of stages which are bled. For practical reasons this number is, however, limited to from two to four. In many cases this number is sufficient even for higher pressures. If the feed-water is heated by steam bled at three different stages, the gain derived from any additional stage is comparatively slight. With five stages the theoretical maximum possible is very closely approached, although there are other advantages besides the increase in efficiency to be obtained by bleeding many stages. Fig. 11 shows the increase in efficiency obtainable by stage heating for different temperatures and different steam pressures.

The author next discusses the subject of high-pressure back pressure and sets forth the advantages of having high-pressure steam in prime movers exhaust against back pressure, which latter means exhaust pressure higher than that of the atmosphere. Among other things, he states that the combination of high pressure and back pressure offers new possibilities to combined power and heating.

The Allegheny Steam Heating Company is building a plant in which there will be installed two boilers of 3000 hp. each. If these boilers were to be built to operate at 800 lb. pressure with 200 deg. of superheat, and the steam used in a turbine expanding to a high back pressure, considerable power would be obtained without a material increase in the amount of coal burned. If the boilers were operated at 200 per cent rating and the steam from the turbines exhausted at 100 lb. back pressure into the heating lines, the amount of power which would be generated would be about 22,000 hp., while the additional amount of coal burned would hardly be noticeable. (B. N. Broido, Chief Engr., Industrial Department, The Superheater Co., New York, N. Y., Mem. A.S.M.E., in *Proceedings of the Engineers' Society of Western Pennyslvania*, vol. 40, no. 9, Dec., 1924, pp. 299–330, 17 figs., and discussion pp. 331–338, figs. 18–20, *gd*)

South African Ash-Handling Plant for Steam-Driven Power-Station Installation

DESCRIPTION of a system of ash handling which has been successfully operated at the generating station of the Rand Power Co. at Vereeniging, and which it is proposed to apply to the power station now being built at Witbank.

One of the factors in determining the selection of the most suitable system of ash handling to be adopted was the ability to obtain comparatively cheap and plentiful native labor, but climatic conditions made also the abolishing of the dust nuisance quite important. Basically the system consists in having the ashes fed continuously from the back drum of the grate through a chute into a trough filled with water and running the whole length of the boiler plant.

The essential features consist of a row of cast-iron tanks filled with water and situated in the boiler-house basement, one tank being provided for each boiler, or alternately for each grate of each boiler. The ash from the boilers is continuously discharged over the back drums of the grates and falls through water-sealed chutes into the tanks where it is quenched, and is then mechanically extracted from the tanks and discharged on a belt conveyor, which in turn delivers the ash to an elevated bunker. One driving mechanism for each row only is required.

Fig. 12 shows a cross-section through a typical Babcock & Wilcox marine-type boiler with chain-grate stoker in the plant, the ash tank (filled to a predetermined level with water), the paddle-wheel extractor and bearing for shaft of same, and the rubber-covered belt conveyor used for conveying the ashes from the boiler house. Coal is continuously fed from the stoker hopper on to the chain-grate stoker, and the ashes are continuously discharged over the dumping bars and fall down the ash chute *A* into a water-filled extractor tank *B*, where they are quenched and fall to the lowest point in the tank. To prevent the ingress of air to the boiler uptake, the ash chute is water-sealed by immersing the discharge mouth of the chute in the water, the chute ending about an inch below the water level in the tank.

Revolving in the tank at a speed of about one revolution a minute, and disposed at an angle of about 45 deg., is a cast-iron wheel *C*, 4 ft. in diameter, the periphery of which is provided with a number of projecting blades; the object of these blades is to lift the ash from the tank bottom and discharge it above the water line, whence the ash falls over the discharging lip of the tank on to the conveyor belt.

Part of the riddlings which fall through the grate near the back drum are largely burned out and of little value, so these are discharged down the chute *D*, into the ash tank and go to the dump with the ashes.

From another figure in the original article it appears that the bearing for the paddle-wheel shaft is carried on a bracket integrally cast with the tank back plate, and as this bearing is long and entirely above water level, and in addition, as the shaft is making only about one revolution a minute, no trouble with wear has been experienced. The paddle wheel, in its rotation, entirely clears the tank, a clearance of $\frac{1}{2}$ in. being allowed between the wheel and the inclined face of the tank, and a similar one on the periphery of the wheel blades.

Working under these conditions and at the extremely low speeds necessary, it will readily be recognized that wear in any part of

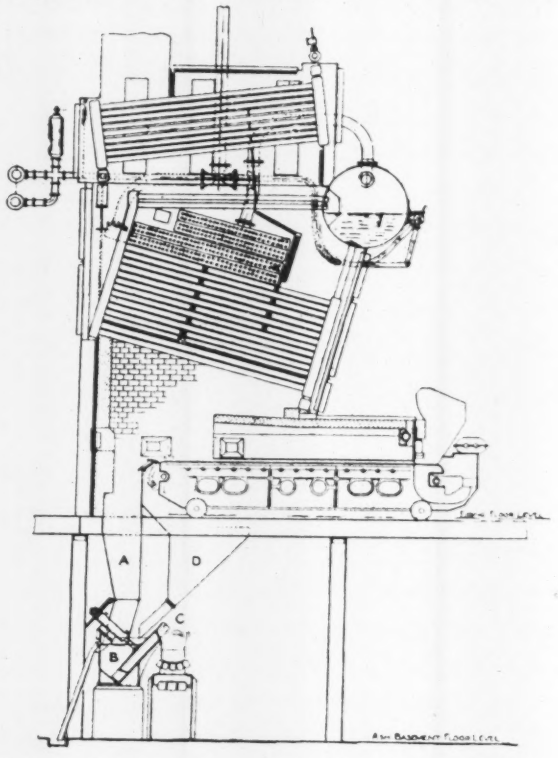


FIG. 12 GENERAL LAYOUT OF THE ASH-HANDLING PLANT AT THE RAND POWER COMPANY'S GENERATING STATION AT VEREENIGING

the extractor mechanism is negligibly small and maintenance is reduced to a minimum.

Between the discharging lip of the tank and the water level, a freeboard of about 6 in. is provided, in order to allow much of the water entrained in the ash to drain back into the tank. An emergency door is provided in the back of the tank to allow for clearing, should the tank get choked with ashes when the extractor wheel was inadvertently stopped. To limit the temperature of the water in the tank, cold water is fed continuously in, and the heated water overflows as shown, into a drain trench cut in the basement floor, the amount of water being fed into the tank being regulated to meet the requirements of the heat to be abstracted from the ashes, and to make up the lost water carried to the dump with the saturated ashes. In practice the water temperature in the tank is maintained at about 150 deg. fahr.

A figure in the original article shows the plan of two rows of extractor tanks and also the belt conveyor required for each row. From this it appears that each row contains ten extractor tanks, five boilers each provided with two furnaces constituting the boiler row.

The problem of finding a simple, inexpensive, and reliable method of driving the extractor wheels for a complete row of tanks was not an easy one. The device finally adopted consists of a steel wire rope about $\frac{3}{4}$ in. in diameter and running the whole length of a complete row of extractor tanks. Each end of the rope is connected by a link to a swinging arm pivoting from a foot secured to the con-

crete floor, the lower part of each swinging arm being fitted with a projecting arm carrying a balance weight. The swinging arm at the driving end of the rope is oscillated to and fro by means of a short connecting rod and a crank disk with a crank throw of about 4 in.

When the crank disk is rotated a slow reciprocating motion of about 8 in. is imparted to the driving rope, which is kept in tension by the balance weights on the swinging arms.

As the swinging arms are about 5 ft. in length from the pivots to the point of rope connection, and as the arc swung through is small, the motion of the driving rope is for all practical considerations a straight-line motion.

The line of the reciprocating rope lies on the driving quadrant operating the ratchet wheel on the shaft of the extractor wheel of each tank in a row, and a simple connection is made from the reciprocating rope to each quadrant. The reciprocation of the driving rope thus imparts an intermittent rotation to the extractor wheels of all the tanks in one row, and the rotation of any individual extractor wheel can be stopped if desired, by simply raising the pawl from the ratchet wheel. (A. Hossack in *Journal of the South African Institution of Engineers*, vol. 23, no. 6, Jan., 1925, pp. 336–347, and discussion pp. 347–351, 6 figs., *d*)

PRODUCER GAS AND GAS PRODUCERS (See Fuels and Firing: Influence of Pulsations on Combustion)

PUMPS

Performance of Centrifugal Pumps When Pumping Oils

REPORT of an investigation carried out at the Los Angeles Refinery of the Union Oil Co. of California for the Goulds Manufacturing Co., of Seneca Falls, N. Y. The primary object of the test was to determine the head, capacity, and efficiency of a centrifugal pump as functions of the viscosity of the liquid handled by it. It was also desired to find the effect of a variation in the vacuum in

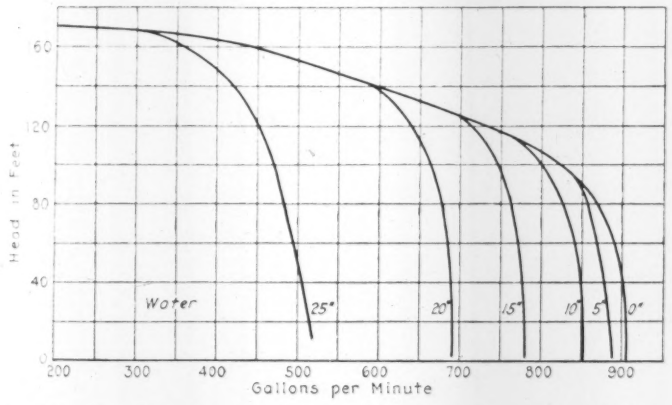


FIG. 13 HEAD-CAPACITY CURVES WITH VARIOUS VACUUMS AT SUCTION INTAKE IN INCHES OF MERCURY (1420 TO 1490 R.P.M.)

each case and to determine the limit of vacuum under which each commodity could be pumped, the physical property here involved being the vapor pressure.

Among other things, attention was given to the calibration of an orifice meter with each of the commodities used and also the determination of the friction losses in the suction pipe between tank and pump. The results are presented in the form of diagrams. As regards head (which was here expressed in feet of the liquid employed) it was found that the head-capacity curve is the same for water, gasoline, and light crude. Then as the viscosity increases the characteristic becomes steeper and steeper, but the rise in the characteristic of the three-stage pump disappears when cold residuum is pumped. This is ascribed to the lack of efficient conversion of kinetic energy in the diffusion vanes when the higher viscosity is reached.

The difference between the single-stage and the three-stage pump is not especially significant. It seems that the capacity of the single-stage pump falls off somewhat less than that of the three-stage, but as higher capacities give us lower heads on the head-capacity curve, the head falls off more. As this is largely a matter of selecting what is the peak of the efficiency curve and many of the efficiency curves are seen to be quite flat, it is obvious that one cannot be very dogmatic on this. In general, therefore, the four curves in Fig. 9 in the original article, expressing the variation in head or capacity, might apparently be replaced by a single average curve which would indicate the variation in both head and capacity.

A series of figures show the effect of the various vacuums. Fig. 13 gives their general characteristics. These are the actual values as read on the vacuum gage. Since it was not desired to place the curves for the various commodities on one sheet, it was less essential to reduce them to one standard speed. Also on high vacuums the determining factor is the exact value of the pressure at the suction of the impeller, and this depends upon the vacuum-gage reading and the exact value of the rate of discharge. It was decided that any irregularities in the curves due to variations in speed were of less consequence than that we should here have the exact values found in the test. Hence these curves are plotted for values as observed. In most of the cases, if not all, the vacuum was carried to as high a value as would give any reasonable discharge. For instance, with gasoline a vacuum of 15 in. of mercury could be obtained but at a very small capacity, and it was physically impossible to go beyond this point. Thus throttling the suction a trifle more would merely cause the flow to cease without increasing the vacuum any appreciable amount.

It will be seen that heating the residuum makes it possible to carry somewhat less vacuum than when it was cold, due to the change in the vapor pressure. Otherwise the only effect of the heating was to change the viscosity. (Robt. L. Daugherty, Professor of Mechanical and Hydraulic Engineering, California Inst. of Technology in *Bulletin 126 of the Goulds Manufacturing Co.*, Seneca Falls, N. Y., 25 pp., 18 figs., *e*)

RAILROAD ENGINEERING

Use of Otheograph in Rail-Joint Tests

DATA of tests on joints in service reported at a Bureau of Standards meeting. It was found that pebbles placed on a street-car rail more than doubled the stresses set up in the rail when cars ran over it during tests made by the Bureau of Standards. The tests were made by clamping on to the joint an electric telemeter (strain gage) of a type invented by the Bureau of Standards. It was not believed that the otheograph, which has been successfully used in testing railroad locomotives, could be used because of doubts as to the ability of that device to stand the tremendous pounding caused by the flat wheels and corrugated tracks encountered in electric railway service. It was reported that a blow with an 8-lb. sledge made a reading on the otheograph as high as did the heaviest locomotive with which it had been used. (From mimeographed preliminary report of the *Bureau of Standards*, Feb. 20, 1925, *e*)

Comparative Costs of Electric and Steam Operation on the Mountain Divisions of the C. M. & St. P. Ry.

THE electric operation of the Rocky Mountain, Missoula, and Coast divisions of this railway over the mountain ranges in Montana, Idaho, and Washington is of interest because it is by far the most extensive operation of the kind ever undertaken, consisting of a total of 650 miles of main-line railway where all the train movements—freight, passenger, switching, etc.—are performed exclusively by electric power.

A careful study of the cost of operation under steam on the identical districts which were afterward electrified has been made and a comparison of the cost of steam versus electric operation is given in the report. It is based on an actual calculation of all the costs involved in both methods of operation and is believed to reflect accurately the relative cost of each.

From a table given in the original publication it would appear that the net savings from 1916 to 1924 on all the electrified sections amounted to $12,400,007. A series of charts give some of the detailed figures from which this comparison was derived, and tables give a list of operating expenses for each division directly affected without change in power, as well as investment in and carrying charges on the property peculiar to each mode of operation. (Publication by the Chicago, Milwaukee & St. Paul Railway Co., 25 pp., several diagrams, *c*)

D. & H. High-Pressure Locomotive

THE information given in the following paragraphs is supplementary to that published in the March issue of MECHANICAL ENGINEERING, p. 211.

The heating surfaces at the sides of the firebox are composed of 306 tubes expanded into the top and bottom drums. Because of this arrangement the amount of stayed surfaces has been considerably reduced and the surfaces themselves are flat and parallel, with the exception of the small area of the fire-door flange. The drum-and-header type of construction, with the top headers carried forward and securely attached to the boiler barrel, also provides an extremely rigid structure.

Steam is taken from the tops of the forward sections of the steam drums through welded collector pipes suspended from the top of each drum. The upper surface of each of these pipes is perforated with several rows of $^1/_2$-in. holes through which the steam enters, thus distributing the gathering steam over a large water surface and removing a tendency for a local surge of steam to lift the water. These collectors are so effective in service that very little water is removed by a desaturator into which the steam goes after leaving them.

The throttle is of unique design, known as the duplex type. There are two throttles, each operated by a separate lever. What is termed the preliminary throttle is used in starting the locomotive. This takes its steam through a reducing valve which limits the pressure to 300 lb. per sq. in. When the locomotive has been started by the use of this throttle the main throttle may be opened, and the two throttle operating levers automatically interlock so that the continued opening of the main throttle picks up and carries with it the auxiliary throttle lever. Closing the main unlocks the connection between the two levers so that the preliminary valve lever is free to move independently in starting. The throttle valves are of the Chalmers type.

One of the most interesting departures from conventional design is in the frame construction. The main frames terminate just back of the cylinders and the front rails are replaced by a combined saddle, frame, and front-deck casting of steel. The high-pressure cylinder is secured on the right side and the low-pressure cylinder on the left side. The smokebox saddle fit is made in a separate casting, which is bolted to the top of the main casting. This permits the removal of the boiler without disturbing either the smokebox saddle fit or the main frame splice.

The only steam passage cored through the steel saddle casting leads from the exhaust passage of the low-pressure cylinder to the front face of the saddle casting on the center line of the locomotive. From this point an outside exhaust pipe carries the steam forward and up into the exhaust nozzle in the smokebox. The center line of the exhaust nozzle is located $65^1/_2$ in. forward of the transverse center line through the cylinders. (*Railway Age*, vol. 78, no. 6, Feb. 7, 1925, pp. 353–357, 5 figs., *d*)

TESTING AND MEASUREMENTS (See also Railroad Engineering: Use of Otheograph in Rail-Joint Tests)

Brinell Hardness Numbers

THE increasing use of the Brinell indentation test for hardness has created a need for a convenient table for converting the diameter of Brinell impressions to Brinell hardness numbers.

Reasoning from an extension of Hertz's theory, Brinell defined the Brinell hardness number (B.H.N.) as P/A, where P is the load impressed on the material through a hardened steel ball and A is the area of the spherical indentation produced. Assuming that this indentation is a portion of a sphere of the same diameter (D) as the indenting ball, the area A may be computed from the diameter (d) of the impression by the formula

$$A = \frac{\pi D^2}{2}\left(1 - \sqrt{1 - \left(\frac{d}{D}\right)^2}\right)$$

so that

$$\text{B.H.N.} = \frac{2P}{\pi D^2\left(1 - \sqrt{1 - \left(\frac{d}{D}\right)^2}\right)} = \frac{2P}{\pi d^2}\left(1 + \sqrt{1 - \left(\frac{d}{D}\right)^2}\right)$$

Experience has shown that the Brinell hardness numbers so computed vary with the applied load (P) and the diameter (D) of the indenting ball. The test has therefore been standardized by prescribing a standard 10-mm. ball and two standard loads: 3000 kg. for hard materials (such as steel) and 500 kg. for soft materials (such as copper, brass, duralumin, etc.). The 3000-kg. B.H.N. will, in general, not be the same as the 500-kg. B.H.N., nor can any general conversion table, applicable to all materials, be given to convert the one into the other. Similarly, where because of convenience or necessity, as in the case of thin specimens, a smaller ball and smaller load are used (Baby Brinell, Rockwell), the hardness numbers obtained cannot be converted by any general conversion table to either 3000- or 500-kg. B.H.N. For this reason, in reporting the B.H.N. the load should always be stated, as well as the diameter of the ball if it differs from the standard 10 mm.

The table included in the publication here abstracted gives the values of both the 3000- and 500-kg. B.H.N. with the standard 10-mm. ball for diameters of impression varying from 2.00 to 6.99 mm. by steps of 0.01 mm.

For a partial bibliography of hardness testing, reference is made to Bureau of Standards Technologic Paper No. 11, and for methods of testing, to Proc. A.S.T.M., 1924, Report of Committee E1 on methods of testing. (Prepared by the Engineering Mechanics Section of the Bureau of Standards, *Miscellaneous Publication of the Bureau of Standards*, no. 62, Dec. 17, 1924, *p*)

THERMODYNAMICS

Influence of Cooling a Gaseous Fluid Previous to Its Compression

THE author gives formulas for the work required for compressing a gas with and without precooling and compares the quantities of energy consumed in each of these two cases. He shows that in order that T', or the energy required to compress precooled gas, be smaller than T, or the energy required to compress the same gas without precooling, it is necessary that

$$\alpha(\mu - 1) < \beta(\lambda - 1) \text{ or } \mu < 1 + \frac{\beta}{\alpha}(\lambda - 1) \text{ or}$$

$$t_0 > \theta_0 > \frac{t_0}{1 + \frac{\beta}{\alpha}(\lambda - 1)}$$

where α is the practical efficiency of compression as compared with the theoretical adiabatic compression; μ not stated; β the practical efficiency of cooling with reference to the minimum theoretical work in accordance with the Carnot cycle; λ is determined from the formula

$$\left(\frac{p_1}{p_0}\right)^{\frac{\gamma - 1}{\gamma}} = \lambda$$

and γ is the ratio of the two specific heats; θ_0 is the temperature to which the gas has been cooled from its initial temperature of t_0.

There is always a series of values of θ_0 which satisfy these inequalities, which is due to the fact that λ is assumed to be greater than unity, that θ_0 which is always greater than zero is seldom less than 0.60, and that finally α, when the isothermal efficiency is 0.63, varies between 0.75 and 0.80 when the ratio $p_1 : p_0$ varies from 2 to 5. The author proceeds to derive expressions for the best temperature to which gas should be precooled before compression and shows that as regards the expenditure of energy the employment of the best temperature of precooling produces exactly the same economy as compression in two stages with an intermediate stage at the most favorable pressure $\sqrt{p_0 p_1}$.

It would appear from this that in large compressor installations, such as those where gases are subjected to catalytic synthesis under pressure and where very powerful compressors are used with five to six stages and the expenditure of power is very high, the artificial cooling of the gas at all stages by a *single* refrigerating machine would give economies of power equivalent to doubling the stages of all the compressors, or for the same expenditure of energy would permit reducing by half the number of stages of all the compressors. (Georges Patart in *Comptes Rendus des Séances de l'Académie des Sciences*, vol. 180, no. 2, Jan. 12, 1925, pp. 120–122, *mtpA*)

Refractories for Oil-Burning Boilers[1]

BY THOMAS S. CURTIS,[2] LOS ANGELES, CAL.

FIREBRICK fail to give their normal or expected life in oil-fired boilers usually for a combination of reasons, two of which are recognized and appreciated. First, there is a tendency to soften or squat under loads, sometimes at temperatures hundreds of degrees below the actual melting point. This failure is brought about by an incipient vitrification of the brick body, that is, a tendency to become stony or glassy. When the glass forms the body of the brick may partake of the properties of asphaltum; that is to say, it is rigid at low temperatures, but with increasing temperature, it becomes plastic and deforms.

The second cause of failure is really an outgrowth of the first, namely, spalling, cracking, and breaking off in pieces. In the observation of a considerable number of installations, the author has yet to see an actual case of failure due to spalling without its predecessor, vitrification, having made its appearance. In other words, he seriously questions the value of the so-called "spall test" commonly used to demonstrate the suitability of a given brick from the standpoint of its resistance to spalling.

With increasing size of units, the dead load on the refractories of course has increased proportionately. Any change in volume in the brick as a unit naturally manifests itself in a warpage, bulging, or leaning of the wall, as the case may be. As the size of the boiler increases, this weakness becomes more and more a real hazard.

As previously indicated, when the failure of the brick has reached the stage of softening or vitrification, the first time the unit is cooled down, cracking and spalling commence. This tendency is usually exaggerated by the boiler tender, who may leave the drafts open to expedite the cooling of the setting. Incidentally the author does not know of any other industry or type of application using refractories where the effects of sudden chilling are so completely ignored as in that of boilers.

When the boiler is again put on the line, a third influence of destruction is brought to bear, namely, the throb or pulsation of the oil flame as a result of the series of miniature explosions by which its combustion is manifested. This throb or pulsation will rock or vibrate a wall weighing many tons as though it were made of paper.

Therefore it is seen what an excellent chance the pieces which have been cracked by the sudden chilling, have to be shaken out of the wall. As a matter of fact, the entire structure is weakened by this vibration.

Let us now see if there is any way in which this destruction can be retarded.

The old question: "What temperature will your brick stand?" has gradually given way under our persistent and well-meaning campaign of education and investigation to: "What will your brick stand at such and such a temperature?"

To illustrate: An excellent brick will bear a load of 25 lb. per sq. in. at 3000 deg. fahr. with not more than 10 per cent deformation, provided the atmosphere is neutral or oxidizing. The melting point of that brick would be 250 deg. higher at least, but the melting point is of small consequence. But if we permit the pores of that same brick to become saturated with reducing vapors at 3000 deg. it will not bear its own weight without squatting, let alone supporting a load—as it must in a furnace wall.

Now the value of a clay brick, in terms of load-bearing resistance, ability to maintain its porosity even at high temperatures under oxidizing conditions so that it shall not spall, and, in short in terms of general stability under high temperature, is measured, first, in terms of the purity of the clay from which it is made, and secondly, in terms of the composition of that clay.

The ideal clay refractory would be dense kaolin, that is, one molecule of alumina to two of silica. But the two oxides must be combined as clay substance. There must not be more than 5 per cent of impurities, and of these not more than one-half of one per cent shall be free or uncombined.

When the company with which the author is associated undertook the investigation of California clays some four years ago, it was found that a number of them had the desired chemical composition in the pure clay substance, but that a hopeless variation in quality existed because of the inclusion of impurities in a free state. After considerable research work, it was found possible and commercially practicable to refine these clays by a special method of concentration which removed the free impurities and delivered a pure clay substance of definite and controllable composition. The result of this early work is the present Argon AA brick, of which several million are now in use in California plants.

However, the best clay brick that can be made, without regard for cost, is subject to drastic change in volume under intense reduction. Especially is this true as the alumina content increases, although with an increase in the alumina content the melting point is sharply raised.

Having learned that a good clay brick is well able to bear the weight at the temperatures employed in modern boilers if the atmosphere be oxidizing or neutral, and being unable to upset the practice which makes this atmosphere impossible of attainment in a boiler setting, we have done the next best thing; that is, prevent the reducing gases from entering the pores of the brick.

The value of glazes on brickwork has long been felt but scarcely understood. If a glaze be fluid, it will cut back and seriously erode the brick, causing as much damage as vitrification and spalling. If it be too refractory, the reducing gases will have started their work before the glaze seals the pores.

In an exhaustive effort to obtain a glaze that would be a happy medium, our research laboratory found what it wanted in sillimanite, which our company was probably the first to produce on a commercial scale in the electric furnace. The beautiful core of pure crystals coming from the furnace ingots is crushed, ground, and purified for shipment to an eastern manufacturer of spark plugs. The surrounding sheath of sillimanite, which averages better than 90 per cent sillimanite compound, is broken and ground, to be purified and made into a paint which, when applied to the furnace wall and heated, becomes converted into millions of tiny sillimanite crystals surrounded by a thin film of glass which closes the pores between the crystals and effectually seals the pores in the exposed face of the brick.

We have thus provided a brick that will easily bear the load at the temperature employed if it is used out of contact with reducing vapors, and we have so sealed its surface pores that these vapors cannot penetrate it. We have at the same time retained that porosity and resiliency so necessary to withstand the shock of sudden changes in temperature.

There is but one feature we have overlooked, that is, the vibration and pounding that comes from the combustion of the fuel. In "Tylox," however, we have provided a continuous mechanical interlocking bond between the shell of the boiler setting and the innermost face of the refractory lining. We have thus threefold insurance against change in volume which ordinarily results in leaning walls. Even though the glaze does not completely protect the brick from penetration of reducing gases, we have tied the wall to the cold, strong, outside support.

The suggested solution to the problem, therefore, becomes: The selection of a brick of sufficient load-bearing ability at the temperature used to remain rigid; to have in the body composition mere traces of free impurities so that gradual vitrification within the body may be avoided or retarded; to coat the face of the brick with a refractory crystalline glaze that will seal the pores but will not drip, cut back, and hasten destruction; and, lastly, to tie the face of the wall to the outer support so that the effects of vibration are minimized.

That is our specification. If air cooling is desired, we favor it highly since, if the preheated air is used for combustion, no loss in efficiency is suffered, and the brickwork is certainly helped out tremendously. But no amount of air cooling will make possible the elimination of any one of the three requisites just enumerated, so long as we use clay refractories.

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[1] Presented before Los Angeles Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Los Angeles, Cal., January 24, 1925. Abridged.

[2] Director of Research, The Vitrefrax Company.

Engineering and Industrial Standardization

Proposed Standards for Commercial Forms

THE efforts to establish standards for invoice, inquiry and purchase-order forms date from 1919, when four hundred and seventeen national associations were invited to a conference in Philadelphia, Pa., to discuss this subject. This first conference was called by the National Association of Purchasing Agents for the purpose of devising a standard invoice form which would fit the needs of the greatest number of business concerns with the least disturbance to existing practices.

Following the conference in Philadelphia the standard forms entered a development stage, and due largely to the efforts of the Railway Accounting Officers' Association, the National Association of Purchasing Agents, the National Association of Cost Accountants, and Division VI, Purchases and Stores, of the American Railway Association, there was evolved a set of forms which attained a widespread use throughout the country. To assist in bringing about the universal adoption of standard forms these associations took advantage of the services offered by the Department of Commerce to aid American industry in its efforts to eliminate waste. Therefore the National Association of Purchasing Agents requested the Division of Simplified Practice of the Department of Commerce to call a conference of all interests in an effort to have them formerly approve the recommended standard forms for use throughout business generally.

This conference was held in Washington, D. C., on January 14, 1925. The meeting was called to order by R. M. Hydson, chief of the Division of Simplified Practice of the Department of Commerce. He outlined the purpose for which the conference was assembled, and the methods used by his division in assisting the various industries to eliminate waste occurring from over-diversification of product and unnecessary duplication of business procedure. There were 56 persons present.

The standard invoice form proposed by the above-mentioned associations was presented to the conference first and, after a long and spirited discussion, it was adopted in the form presented. This form is to be printed on paper $8^1/_2 \times 7$ in. and $8^1/_2 \times 14$ in. At the opening of the afternoon session the Chairman presented the proposed standard purchase-order form. It was explained by W. L. Chandler, Secretary of the N.A.P.A., that due to the wide divergence of requirements in various business, the committee felt that all it could do was to divide this form into zones to show the desirable arrangement of information. He called attention to the sizes of the various zones, and discussed the reasons for the proposed dimensions for the purchase order forms. He called attention to the close similarity between the purchase-order form and the inquiry form, and moved the adoption of these two forms as standards. A discussion of the sizes of the various forms led to the proposal of an amendment, to the effect that the sizes approved for the invoice form apply to the purchase-order and inquiry forms as well. These standard sizes are: $8^1/_2 \times 7$ in., $8^1/_2 \times 11$ in. and $8^1/_2 \times 14$ in. The original motion and the amendment were unanimously passed.

The conference adopted a motion urging the immediate acceptance of these standard forms. Copies of them may be obtained upon application to W. L. Chandler, Secretary, National Association of Purchasing Agents, Woolworth Building, New York, N. Y.

Standardization and the Problems of Distribution

OWING to the widespread misunderstanding of what takes place in distribution and the necessity for detailed studies of their problems by distributers themselves, the Board of Directors of the Chamber of Commerce of the United States authorized the calling of a national distribution conference.

This took place in Washington on January 14 and 15, 1925. Although called by its president, R. F. Grant, the Conference was under the auspices of the Chamber and was composed of about 200 business men, representative of agriculture, manufacture, wholesale and retail distribution, without regard to whether or not they were members of the Chamber.

Secretary Hoover delivered an inspiring address at the opening session in which he answered in the affirmative the question: "Can we reduce the margin between our farmer and manufacturing producers on one side, and our consumers on the other?" In the elimination of waste, he claimed, lie the possibilities of this accomplishment, and in the list of fifteen sources of this waste he included (*a*) waste from lack of standards of quality and grades, and (*b*) waste arising from the unnecessary multiplication of terms, sizes, varieties.

Safety Code for Walkway Surfaces

AS previously announced, a Sectional Committee on a Safety Code for Walkway Surfaces has been organized under the procedure of the American Engineering Standards Committee. The Sponsors for this Committee are the American Institute of Architects and the American Society of Safety Engineers, Engineering Division of the N. S. C. and its officers are Lucian W. Chaney, chairman, and H. Weaver Mowery, secretary. The personnel is now complete, consisting of 43 members representing 36 organizations.

The scope of this project has now been definitely defined as follows:

"Pertinent walkway surface characteristics; including resistance to slipping, freedom from tripping hazard, durability, flammability; insulation, as well as non-slip characteristics, of walkway surfaces around electrical apparatus; general and maintenance requirements. For use in apartment houses, factories and other work places, office buildings, hospitals, hotels and restaurants, railway cars, railway stations and train platforms, schools and theaters; for elevator floors, elevator landings, corridor floors and door thresholds, ramps, runway floors, stair and fire-escape treads and landings, sidewalks, accessories to buildings, including coalhole covers, sidewalk doors and gratings, floors and platforms around moving machinery and electrical apparatus."

Cordage Investigation

RECENT experimental work at the U. S. Bureau of Standards has shown that the physical properties of rope are greatly influenced by variations in the physical make-up, including twist, size of yarn, lay, and protective coatings. The study of the effects of these variables has been made possible by the design and construction of two machines, one of which tests the individual yarns, and the other, the completed rope.

The preliminary work on the yarn-testing machine has been completed, and a standard method of test adopted. Research is under way on the sensitivity of the machine to slight changes in the above-mentioned variables. The effect upon the yarn of varying amounts of oil and acids will also be studied.

The machine for testing the completed rope is in process of construction, and when finished the standardization of the variables will be undertaken.

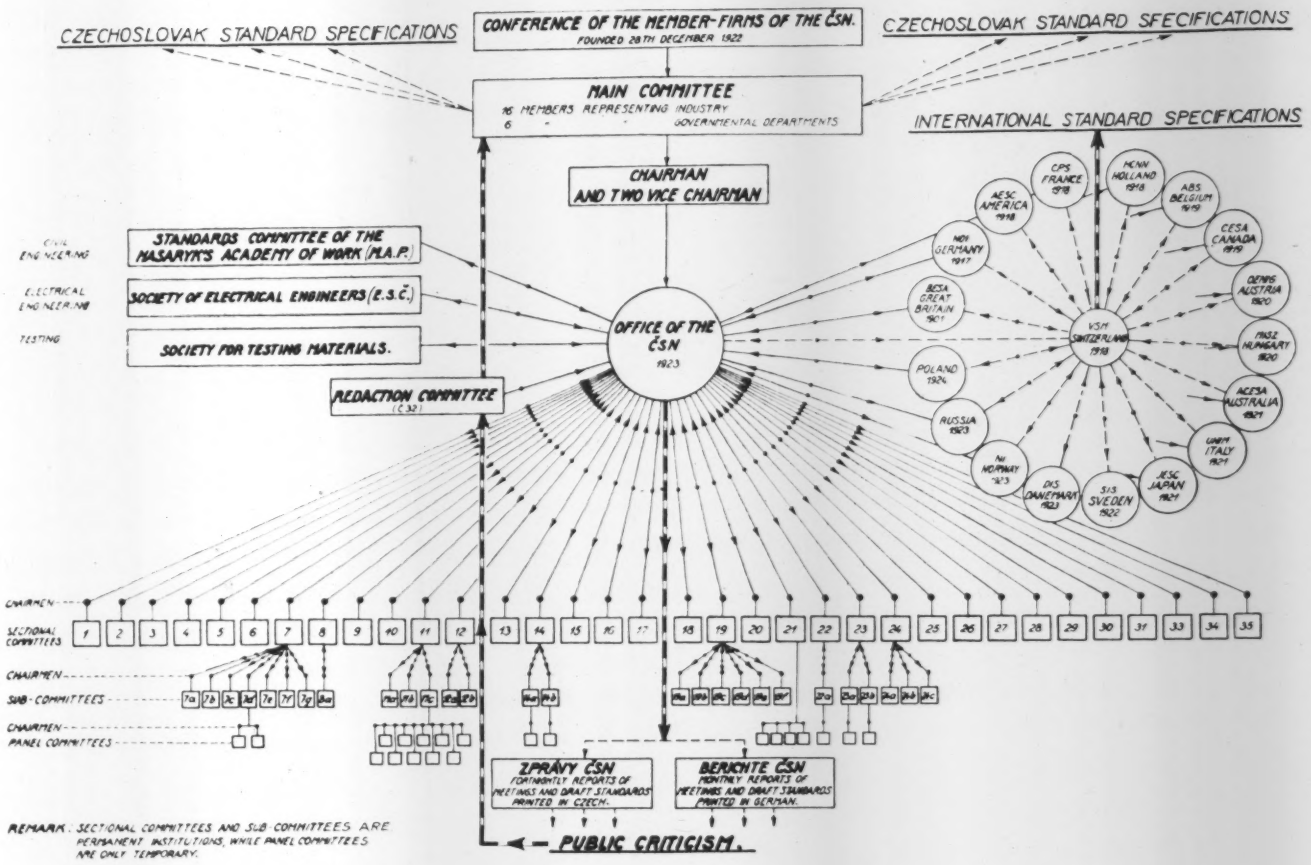
A complete study of the microchemical analysis and identification of the common cordage fibers is likewise being made. Preliminary results indicate that this will ultimately prove an important aid in routine fiber identification.

A machine has also been constructed for subjecting sash cord to a service test. Sash cord is the most important of the various kinds of soft-fiber cordage. Preliminary tests on this machine have just begun, and it is proposed to test a large number of samples with the object of standardizing the machine and methods of test.

Summarizing the work, a combination of tests is being devised which, when completed, will make possible a complete analysis, both physical and chemical, of cordage, and will set definite standards for its working value.

ČESKOSLOVENSKÁ NORMALISAČNÍ SPOLEČNOST.
(CZECHOSLOVAK ENGINEERING STANDARDS ASSOCIATION)
MIKULAŠSKÁ 28, PRAGUE.

DIAGRAM OF ORGANIZATION OF THE NATIONAL STANDARDIZATION IN CZECHOSLOVAKIA. DECEMBER 1924.


CZECHOSLOVAK STANDARD SPECIFICATIONS
CONFERENCE OF THE MEMBER-FIRMS OF THE ČSN.
FOUNDED 28TH DECEMBER 1922
CZECHOSLOVAK STANDARD SPECIFICATIONS
MAIN COMMITTEE
16 MEMBERS REPRESENTING INDUSTRY
6 " " GOVERNMENTAL DEPARTMENTS
INTERNATIONAL STANDARD SPECIFICATIONS
CHAIRMAN AND TWO VICE CHAIRMAN
CIVIL ENGINEERING
STANDARDS COMMITTEE OF THE MASARYK'S ACADEMY OF WORK (M.A.P.)
ELECTRICAL ENGINEERING
SOCIETY OF ELECTRICAL ENGINEERS (S.E.Č.)
TESTING
SOCIETY FOR TESTING MATERIALS.
REDACTION COMMITTEE (C 32)
OFFICE OF THE ČSN 1923
AESC AMERICA 1918
CPS FRANCE 1918
MCNN HOLLAND 1918
ABS BELGIUM 1919
NOI GERMANY 1917
CESA CANADA 1919
BESA GREAT BRITAIN 1901
DENOS AUSTRIA 1920
VSM SWITZERLAND 1918
MASZ HUNGARY 1920
POLAND 1924
ACESA AUSTRALIA 1921
RUSSIA 1923
UNIM ITALY 1921
NI NORWAY 1923
DIS DANEMARK 1923
SIS SVERIGE 1922
JESC JAPAN 1921
CHAIRMEN
SECTIONAL COMMITTEES 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 33 34 35
CHAIRMEN
SUB-COMMITTEES 7a 7b 7c 7d 7e 7f 7g 8a 10a 10b 10c 10d 10e 14a 14b 18a 18b 18c 18d 18e 18f 23a 23b 23c 24a 24b 24c
CHAIRMEN
PANEL COMMITTEES
ZPRÁVY ČSN
FORTNIGHTLY REPORTS OF MEETINGS AND DRAFT STANDARDS PRINTED IN CZECH.
BERICHTE ČSN
MONTHLY REPORTS OF MEETINGS AND DRAFT STANDARDS PRINTED IN GERMAN.
REMARK. SECTIONAL COMMITTEES AND SUB-COMMITTEES ARE PERMANENT INSTITUTIONS, WHILE PANEL COMMITTEES ARE ONLY TEMPORARY.
PUBLIC CRITICISM.

SECTIONAL COMMITTEES AND SUB-COMMITTEES OF THE ČSN.

C 1. SIZES OF PAPERS.
C 2. TECHNICAL DRAWINGS.
C 3. BOILERS. SPECIFICATIONS OF MATERIALS.
C 4. BOILERS. DIMENSIONS AND STRUCTURE.
C 5. BOILERS. RULES FOR MANUFACTURERS.
C 6. STEAM TURBINES.
C 7. ROLLED SECTIONS FOR STRUCTURAL PURPOSES.
SC 7a. BEAMS AND CHANNELS.
SC 7b. ANGLES AND T BARS.
SC 7c. RAILS.
SC 7d. VARIOUS ROLLED SECTIONS.
SC 7e. RIVETS.
SC 7f. TRAMWAY RAILS.
SC 7g. ROLLED SECTIONS. MECHANICAL PROPERTIES, TOLERANCES, WEIGHTS.
C 8. QUALITIES OF IRON AND STEEL.
C 9. ROLLED SECTIONS RECTANGULAR, ROUND, HEXAGONAL ETC.
C 10. AUTOMOBILES.
C 11. AIRCRAFT.
SC 11a. AIRCRAFT. BODIES.
SC 11b. AIRCRAFT. ENGINES.
SC 11c. AIRCRAFT. MATERIALS.
C 12. SYSTEM OF LIMIT GAUGES.
SC 12a. LIMIT GAUGES FOR MECHANICAL ENGINEERING.
SC 12b. LIMIT GAUGES FOR FINE MECHANICS.
C 13. TUBES. GENERAL SUBJECTS.
C 14. SHAFTING. GENERAL SUBJECTS.
SC 14a. SHAFTING. NON-ROTATING PARTS.
SC 14b. SHAFTING. ROTATING PARTS.
C 15. SCREW-THREADS.
C 16. CHAINS.
C 17. WIRE-ROPES.
C 18. SCREWS. NUTS. WIDTH OF JAWS OF SPANNERS.
C 19. ELEMENTS /PARTS/ OF GENERAL ENGINEERING.
SC 19a. SERIES OF PREFERENCE NUMBERS.
SC 19b. KEYS AND KEYWAYS. PLUGS. TAPERED PINS. SPLIT PINS.
SC 19c. HAND WHEELS. HANDLES. HAND LEVERS ETC.
SC 19d. CONES. SQUARE HEADS. MACHINE TOOL FIXTURES.
SC 19e. ANCHOR BOLTS AND PLATES.
SC 19f. BEARINGS. BEARING BUSHES. LUBRICATING RINGS AND CUPS.
C 20. PLOUGH-COLTERS.
C 21. TESTING METHODS FOR CAST IRON.
C 22. CAST IRON SEWAGE PIPES.
C 22a. VALVES AND ACCESSORIES FOR DTO.
C 23. CAST IRON SEWAGE PIPES.
SC 23a. VALVES AND ACCESSORIES FOR DTO.
SC 23b. VALVES AND ACCESSORIES FOR CAST IRON GAS PIPES.
C 24. WROUGHT IRON TUBES.
SC 24a. FLANGES, SCREW-THREADS AND FITTINGS FOR DTO.
SC 24b. VALVES AND ACCESSORIES FOR STEAM PIPES.
SC 24c. VALVES AND ACCESSORIES FOR STEAM AND WATER HEATING PLANTS.
C 25. WROUGHT IRON SEAMLESS AND PRECISION TUBES. TUBES FOR SPECIAL PURPOSES.
C 26. COPPER AND BRASS TUBES AND FITTINGS.
C 27. LEAD AND TIN TUBES AND FITTINGS.
C 28. GLAZED WARE PIPES.
C 29. CONCRETE PIPES.
C 30. MATHEMATICAL AND PHYSICAL SYMBOLS.
C 31. SYSTEMS OF IDENTIFYING STANDARDIZED PARTS.
C 32. REDACTION COMMITTEE.
C 33. ACCURACY AND METHODS OF MEASUREMENTS OF MACHINE TOOLS.
C 34. WOOD PARQUETS.
C 35. VARNISHES AND PAINTS.

PRAGUE, DEC. 1924.

Correspondence

CONTRIBUTIONS to the Correspondence Department of "Mechanical Engineering" are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

Method of Indicating Tolerance Limits on Drawings

To the Editor:

We are desirous of obtaining information on methods of dimensioning drawings, with particular reference to detail machining dimensions.

Example—A fitting 2 1/8 in. nominal diameter:

Desired clearance between shaft and hole	0.003 in.
Minimum clearance	0.002 in.
Tolerance on shaft	0.001 in.
Tolerance on hole	0.002 in.

	Hole	Shaft
No. 1	$\frac{2.127}{2.125}$	$\frac{2.123}{2.122}$
No. 2	$2.125 \; {}^{+\;0.002}_{-\;0.000}$	$2.122 \; {}^{+\;0.001}_{-\;0.000}$
No. 3	2.126 ± 0.001	2.1225 ± 0.0005

Other methods making use of fractional dimensions with tolerances in decimals may be considered, but the three examples given will serve to illustrate the point in question.

It may be said for No. 1 that it has the advantage of simplicity, having only the maximum and minimum to deal with both from the drafting-room and machine-shop point of view, and the disadvantage of no intimation to the machine operator as to the desired fit clearance between the two parts.

No. 2 may be considered as having the advantage of a mark for the operator to aim at that will give the desired clearance, and the disadvantage in aiming for the desired dimension of slipping slightly in this case below the aim; also requiring slightly more effort in the drafting room than No. 1.

No. 3 may be said to have the advantage of giving the machine operator an equal plus and minus latitude, and the disadvantage of not so nearly informing the operator the desired fit clearance; also the disadvantage of greater effort in the drafting room than on No. 1.

J. W. B. Pearce.[1]

To The Editor:

Two different methods of indicating tolerance limits have been employed in our drafting room. The latest one (see Fig. 1) was worked out by the writer in 1923 and has been in use for about one year, with much more satisfaction than when drawings were dimensioned as indicated under the column "Present Method," which was only one of several other methods previously used.

What prompted us to change our method of dimensioning was that with only the maximum and minimum tolerance given, the men would invariably ask which of the two we should rather have them work to. Some men, of course, would mentally divide up these limits and make their work half-way between, which in most cases would be satisfactory. When only minus or plus tolerances are desired, however, this cannot be indicated as well when giving dimensions in this manner. With the dimensioning changed as shown in the second column, it is plainly indicated which dimension is wanted; and it is also possible to indicate properly the plus or minus tolerances, if both are given, when they are of different amounts. It should also be noted that this method of dimensioning eliminates the writing of quite a number of figures, thus making the drawings plainer and easier to read.

[1] Asst. Engr., Spicer Mfg. Co., South Plainfield, N. J.

As stated above, we have used this method of dimensioning for approximately one year, and from personal inquiries among the men in the shop using these drawings we have not found one who did not state that he considered this system the best that he had ever used. For this reason the writer is submitting the scheme to the

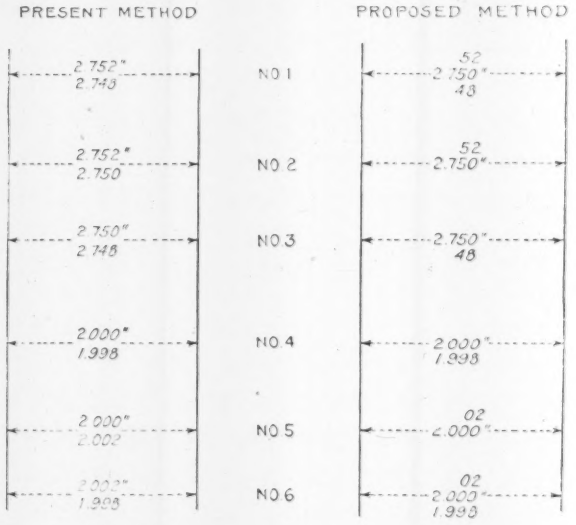


Fig. 1 Proposed Improved Method of Indicating Tolerances on Drawings

Society, thinking that if such matters are taken up at its meetings, others might want to consider the adoption of this method of dimensioning, and probably benefit by it as we have.

M. E. Widell.[1]

Cincinnati, Ohio.

Circular, Cylindrical, and Spherical Units of Measurement

To the Editor:

Mr. Spitzglass' suggestion for the use of circular units of measurement is interesting and deserves careful consideration. It is, however, impossible to state a priori whether the suggestion is practical or not. An area or volume unit to be practical must serve as a basis for conveniently comparing areas or volumes in a wide field of practice. The square inch (foot, mile, etc.) and cubic inch (foot, mile, etc.) came into use because in the early history of measurement rectangular areas and parallelopipedal volumes were those in most common use. It was not until an electrical industry grew up in which cross-sections almost exclusively circular had to be compared frequently that it became convenient to introduce the circular mil as a unit of area. Here it is more convenient to divide by π in the relatively few cases where rectangular wires are used than to multiply by π in every one of the thousands of cases where circular wires are used.

In another way the circumstances were very favorable to the introduction of such a unit. Volume computations were relatively infrequent so that the relation of the area unit to the volume unit was of minor importance. The third dimension of the wires was only involved in the novel combination Length × Area, for which no unit existed, and it was easy and logical to introduce the mixed unit, the "mil-foot."

In mechanics the problem is less simple. The area unit is bound up with the volume unit so that here the introduction of the circular inch as the area unit can only be an advantage if at the same time we introduce the cylindrical inch as the unit of volume.

[1] Chief Designer, American Car Co.

In industry today, as Mr. Spitzglass has pointed out, circular areas are mechanically related to cylindrical volumes sufficiently frequently to make it seem quite possible that this combination of area and volume units would prove very useful. I do not, however, agree with him that their introduction "would not involve any difficulties even at the start." Before an engineer computes, and while he computes, and after he computes, he must hold in his mind estimates of the relative magnitude of the quantities he is dealing with. He must think his quantities, and this kind of thinking is no small part of his mental effort. It is none too easy for one trained to think "pounds per square inch" (as an American) to think "tons per square inch," as the English do, with a long ton of 2240 pounds. The introduction of these two units would add two more to the already numerous group of translations of this kind which must be constantly held in mind.

From Mr. Spitzglass' presentation it seems to me possible and even probable—but by no means certain—that the gain would be worth the cost in the case of the circular and cylindrical units. It can only be decided by the experience of practical work with the system. I can see no real reason for the introduction of the spherical unit. If we are to have cylindrical units, the sphere is easily calculated as $^2/_3\ D^3$ cylindrical inches, and a sphere is not a sufficiently common shape to make it worth while to introduce another unit to escape multiplying by $^2/_3$.

L. B. TUCKERMAN.[1]

Washington, D. C.

Equipment Used for Aerial Surveying

TO THE EDITOR:

The paper on Aerial Surveying by Ernest Robinson in the March issue of MECHANICAL ENGINEERING contains a reference to stereocomparators worked out in this country.

Having participated intimately in the development of these instruments by the engineering staff of the Arthur Brock, Jr., Tool & Manufacturing Works of Philadelphia, the writer would like to state that the limit of contour distinction to 20-ft. intervals attributed to these instruments is erroneous. They will yield as dependable results on 5- and 10-ft. contours as the beautiful instrument developed by Dr. Pulfrich. It all depends upon the base length and the flight elevation of the views that are being examined.

Thus far, the commercial work upon which our own instruments have been used did not require closer intervals than 20 ft., but contours at 5-ft. intervals are well within the scope of the instruments we constructed. In fact, these limits are not of a mechanical nature, but depend on the limits of distance perception by the human eye.

A further and very vital point in the science of aerial topography, not brought out in Mr. Robinson's paper, is that the various instruments of foreign make mentioned by him, as well as those worked out for use with the Brock method, make use of glass-plate negatives and positives to the complete exclusion of film. The latter material may not be objectionable when the photographic mosaic, inaccurate at its best, is the final aim, but for accurate topographic map construction it is worthless. Again, a well-designed focal-plane shutter will not introduce larger errors than those directly attributable to the use of film; they may therefore be used with impunity when mosaics are to be made, and advantage taken of their high efficiency. In connection with aerial topography, however, lens shutters are essential, and accordingly the Brock magazine plate cameras are so equipped.

Fig. 7 of Mr. Robinson's article, its caption, and the descriptive matter referring to it seem to indicate that he expects to produce "fairly accurate contour lines" on paper prints from film negatives with the aid of an ordinary viewing stereoscope, such as shown in this illustration. In the absence of any measuring device whatsoever and in consideration of the errors introduced by the specified materials, it is, to say the least, very improbable that contours so located will be anything more than rough form delineations of the image as seen. Since image-point displacements of only 0.005 in. may correspond to elevation differences of 20 ft. or more, no dependable results can be expected from the combination of the instrument and material there illustrated.

L. J. R. HOLST.[1]

Philadelphia, Pa.

TO THE EDITOR:

Replying to Mr. Holst's comments as applied to the reference made in his paper on Equipment Used for Aerial Surveying, appearing in the March issue of MECHANICAL ENGINEERING, with particular reference to stereocomparators worked out in this country, the writer would say that the objections raised by Mr. Holst to the use of film are in theory correct but are only of purely academic interest. In practice, the errors arising from the use of film are negligible alongside of the residual errors remaining after distortion due to tilt, displacements caused by perspective, and discrepancies in scale have been corrected. No process or device, or combination of devices, has yet been invented that will reduce such errors to zero. Very considerable hand labor and personal judgment are required in the application of any process involving the use of the stereocomparator principle. Hence, as long as human beings are not infallible, maps made by these elaborate processes will still continue to be subject to errors from these sources, as are aerial maps made by less involved processes. The writer, however, should have mentioned that glass-plate negatives and positives were used in stereocomparator instruments as a point of information, but does not consider it a vital one for the reasons above mentioned.

There are a multitude of problems yet to be solved in perfecting apparatus used in the airplanes before the full measure of refinement can be obtained by the use of the stereocomparator method whereby the work can be done at a reasonable cost.

As regards the production of 5- and 10-ft. contours by means of the stereocomparator principle, the writer did not intend to convey the idea that this could not be done. It is no doubt within the range of any of the machines described and also of that of the apparatus Mr. Holst mentioned, but very expensive.

On the other hand, a large number of aerial topographical maps have been made by other methods of plotting in contours, also by using film (which Mr. Holst claims is worthless), for a considerable number of power companies, and while not dead accurate, they have all been within the limit of permissible variation to adequately answer the purposes for which they were intended and at a reasonable cost. The best proof of this statement is repeat contracts.

The United States and Canadian Governments are making consistent use of film in producing high-grade maps, and as yet have found no reason to discard film to return to the plates with which they started years ago; again, they have abandoned the focal-plane shutter for the "between-the-lens" type, which the writer believes needs no further comment as both these governments have made more aerial mosaic and topographical maps than all of the commercial concerns put together.

With respect to the statement that fairly accurate contour lines could be obtained, the remainder of the paragraph should reveal clearly enough to any one that since these lines were obtainable by interpolation and by sketching lines through all points of apparent equal elevation, they would not be any more than rough form delineations of the image as seen, and no greater accuracy than that was attempted to be claimed by the writer.

ERNEST ROBINSON.[2]

New York, N. Y.

"Engineman" or "Engineer"

THE United States Civil Service Commission has adopted the use of the term "engineman" in lieu of "engineer" in advertising examinations for positions, the duties of which are the operation, maintenance, and repair of stationary or moving engines the performance of which does not require a professional education.

In reaching a decision as to the adoption of this usage the Commission wrote to classification experts, engineering societies, and individual personnel men. The opinion was unanimous that the term "engineman" was more suitable and less susceptible to confusion than any other term which might be used.

[1] Engineer Physicist, Bureau of Standards.

[1] Cons. Engr., Brock & Weymouth, Inc. Mem. A.S.M.E.

[2] Vice-Pres., Fairchild Aerial Camera Corp. Mem. A.S.M.E.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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E. D. DREYFUS

PUBLICATION STAFF:

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Muscle Shoals

ONE of the last things Congress did before adjourning early in March was to adopt a resolution asking "that the President create a Muscle Shoals commission of three members comprised of a representative citizen familiar with the Muscle Shoals enterprise, a representative of agriculture versed in the use of commercial fertilizers, and a chemical engineer versed in the methods of air-nitrogen fixation.....to obtain information to determine the best, cheapest, and most available means for the production of nitrates in accordance with the National Defense Act, ...and to ascertain the most favorable conditions under which this property may be leased, primarily to secure nitrates for fertilizers in time of peace and explosives in time of war." The resolution further asked that the commission be requested to lay all information and recommendations before Congress at the beginning of the next session.

Almost a year ago a committee of the American Engineering Council, appointed at the request of Senator Norris, chairman of the Senate Committee on Agriculture, to study the problems relating to the proper development of Muscle Shoals for the best interests of the country, addressed a report to Senator Norris expressing unanimous agreement that no disposition of the plants should be decided upon until a thorough investigation of the whole problem had been made by a disinterested technical commission

In the year that has elapsed since this committee of the A.E.C. made its report, Congress has moved more or less in a circle in the Muscle Shoals matter; but it has finally taken the step which is the first essential to clear thinking on the situation. In the thirty-odd years during which much Congressional time has been wasted in useless discussion and argument about Muscle Shoals, the situation there has kept changing. At no time, however, was it a question for legislators to settle without the most expert technical advice obtainable. Today it is a question of how best to develop and maintain at the Shoals industries of peace whose plants can be converted to war purposes at a moment's notice. As such it becomes a problem of our national defense, and any commission engaged in its investigation must have the best advice of the War Department constantly at hand.

It is superfluous to say that competent, disinterested engineering advice is essential to decision on the eventual disposition of the Shoals property. The hydraulic- and hydroelectric-construction features of its development and the methods of manufacture of electrochemical products are matters requiring the skilled advice of technical men trained in those specialized fields of engineering. Whatever plans the President's commission evolves to present to Congress, they must be entirely sound from an engineering standpoint.

Before a decision can be arrived at as to whether Muscle Shoals is of importance from the point of view of the manufacture of fertilizers, it is essential to determine whether there is a real need today for the type of fertilizer that requires large amounts of electric power for its manufacture, such as, for example, calcium cyanamide. From recent developments it would appear that fertilizers made by straight chemistry and catalyzers may be preferable, and for such fertilizers a plant like Muscle Shoals is, of course, unnecessary. Hence a competent plant chemist and biologist is the key man necessary today for any sound proposition on the manufacture of fertilizers. The Department of Agriculture has done remarkable work in experimenting on plant chemistry, but the problem of just what fertilizers it would be best to manufacture is still an open one.

Congress has demonstrated thoroughly, with its long record of contradictory and deferred actions on Muscle Shoals, that it has been incapable of lucid action on the subject. Such questions as government or private ownership and operation of the project come within the field which the nation expects Congress to decide upon, but sound plans upon which to base their action can be evolved only through the efforts of a commission cognizant of the technical, chemical, and commercial aspects of the situation. Much water has gone under the bridge during the years that Congress has failed to do the one essential thing: namely, appoint a commission to present sound plans for the disposition of Muscle Shoals. And yet, the delay has not been entirely fruitless; for so rapidly does science progress that it is possible for a technical commission to approach the subject today more intelligently than it could have approached it a few years ago. At the time the first Ford offer was made it was specifically stated that fertilizers of a cyanamide type should be made. Nobody would dream of doing it today in the light of the information collected in the last three years on the behavior of calcium cyanamide in soils. The method of manufacture of electrochemical products at the Shoals, once so definitely linked with the tremendous water resources available, is now an open question to be approached with an open mind.

It should be possible for a competent technical commission to lay before Congress at the beginning of its next session definite and reliable information upon which to base action. Hope lies now in the President's commission, in its integrity of purpose and its willingness and ability to surround itself with the best advice that modern science can give it.

Cutting Metals

THE cutting of metals presents many interesting phases, not the least of which is the fact that despite the immense importance of the art to industry, much remains to be learned of the fundamental principles that govern the how and why of cutting-tool action. The classic tests of Taylor resulted in formulas and slide rules which could be used in the shop. Since then the major portion of the work done in this field has dealt with such questions as power requirements for cutting metals, milling-cutter tooth shapes, pressures exerted on cutting tools, etc. These are important details, but many such experiments over a wide range of materials and methods and the expenditure of a tremendous amount of time and money would be required before the results would justify the formulation of any scientific laws governing the performance of cutting tools.

In the last analysis any research must have as its end product the development of laws that may be applied in reducing the cost and increasing the volume of metal cut. Research in cutting metals demands an engineering touch, for the problem is distinctly economical. Although in the zone between engineering and pure science, it must be handled by engineers, but with the close coöperation of physicists and metallurgists.

While the importance of fundamental science in cutting metals has been realized by a few leaders in this country, nevertheless the Institution of Mechanical Engineers of Great Britain is apparently the first organization to take definite steps toward a

research program. In 1920 the Cutting Tool Research Committee of the Institution withdrew its research program conceived on practical lines and set up one on fundamentals which resulted in several interesting papers being presented at a meeting of the Institution held during January. The Institution is to be congratulated on the scope of its program and on the patience with which it has pursued its objective. The results obtained will give a valuable stimulus to the work of the Research Committee on Cutting and Forming Metals set up by the A.S.M.E. in 1923.

French Plans for Improving Industrial Education

A BILL has been introduced, with government support, in the French Chamber of Deputies which places the financial burden of improving the elementary stages of technical and industrial education squarely on the shoulders of employers of labor or those who will profit directly most from any improvement resulting from such education.

According to this bill a special tax is to be laid on all employers of labor with the exception of those employing less than eight men. This tax is to amount to 0.35 per cent of the wages paid. Of this amount 0.30 per cent is to be used for the establishment of industrial, technical, and trade schools and 0.05 per cent for the expenses of administration, but of no part of the funds so collected is to be used to meet general budget deficiencies. Further, employers who already maintain apprentice courses or otherwise carry on the work that the new law provides for will be relieved in part or completely of the payment of the new tax.

From such information as is available in this country, it would appear that the bill has been received in a very friendly manner by the French employers of labor, who apparently feel that by this means and at a cost which will not place any serious burden on industry, they are likely to secure a very much better quality of labor. The only fear which they have had that money so collected would not be used for its proper purpose but would go into the general deficiency fund—seems to have been allayed by certain statements made on behalf of the Herriot government.

It would be interesting to determine for the sake of comparison, if sufficient data are available, how the amount spent on elementary industrial and trade schools in this country in proportion to the total wages paid compares with the 0.35 per cent of wages specified for this purpose by the French government.

Presenting Approved American Standards to Cleveland Manufacturers

CLEVELAND engineers have organized a committee to assist in securing the adoption of American Standards in the industries of Cleveland. On March 2 a meeting of the Cleveland Section of The American Society of Mechanical Engineers was held under the auspices of this committee. Dr. P. G. Agnew, Secretary of the American Engineering Standards Committee, was the principal speaker, and an abstract of his remarks follows. At the close of Dr. Agnew's address, Calvin W. Rice, Secretary of the A.S.M.E., gave a brief account of the visit to Czechoslovakia to attend the International Management Congress during the summer of 1924.

DR. AGNEW'S ADDRESS

THE appointment of the Cleveland Engineering Standards Committee marks a notable step in the development of the national industrial standardization movement in this country.

As has often been pointed out, the first stage in industrial standardization in this country, namely, standardization within the factory, was primarily a development of the latter half of the 19th century. It led to America's foremost position in mass production. Standardization by industrial groups, through technical societies and trade associations, is largely a development of the present century, while inter-industry or true national standardization is a recent development, primarily during and since the war. Through the American Engineering Standards Committee the national standardization movement has now taken definite form and direction; the methods and procedure have been thoroughly tried out for more than five years, and a substantial amount of work has been accomplished. About 70 standards have been approved, and a hundred others are under way.

Among the benefits which a company derives from aligning its work closely with the national standardization movement are: it broadens and regularizes sources of supplies; it broadens the market for manufactured products; it stabilizes business by making it safe for the manufacturer to accumulate stock during periods of slack orders to an extent which would not be safe with an unstandardized product; and products complying with national standards are given a large amount of free advertising through an extensive discussion among organizations, and in the technical and trade press, during the period of the formulation of the standards.

While a company may gain many advantages from following the nationally approved standards in a general way, it is only by an intensive and detailed study of many problems that arise in the complete introduction of standards into practice, that a company can gain the most possible from the movement. That is where the Cleveland Engineering Standards Committee comes in—in assisting the Cleveland industries in making such detailed studies, in organizing for it, and in the local interchange of experience.

The Cleveland Committee is the first local committee of the kind to be organized in any of our great industrial centers, and the many industrial organizations and groups interested in these subjects will watch the development of your work with keen interest.

Increasing competition is bound to result in some form of definite organization for the work in the majority of all but the smallest companies. This has already taken place in Continental Europe, particularly in Germany, Austria, Switzerland, and Czechoslovakia. More than 1000 firms in Germany have such organizations, some of the larger companies having as many as 100 or even 200 members of their staff devoting full time to such activities. The increase in the number of associations and companies affiliated in the membership of the American Engineering Standards Committee is a notable step in this direction.

I believe that the work of your Committee will be along similar lines.

It is worth while to mention a few American standards which should be in extensive use in Cleveland. I shall purposely choose them at random: specifications for cement; fire tests of materials and construction—I understand that your city is revising its building code, and this important work should be brought to the attention of the city authorities, if it has not already been done; a code for the lighting of factories—the most authoritative document of its kind; a safety code for ladders—an example of a detailed standard which should be used as a basis of purchase; a code for lighting of school buildings; specifications for various paving materials; pipe thread; codes for abrasive wheels, for foundries, and for foot and hand presses; specifications for steel pipe; electrical codes; specifications for rails for electric car lines; specifications for carbon-steel and alloy-steel forgings; screw stock; copper and copper wire.

Colonel Peck, the chairman of your Standards Committee, has taken a very important part in the development of two fundamental standards which are applicable to a wide range of the mechanical industry, and it is natural that the first two sub-committees which are appointed deal with these subjects. They are, gages and gaging system, under the chairmanship of Mr. Kartsher; and screw threads, the sub-committee for which is now being appointed.

Your Committee and its sub-committees should be of invaluable service to the local companies, in systematizing their standardization work and correlating it with the standardization movement. This will include many problems, such as the detailed methods of introducing it into drafting rooms, shops, stores, and sales departments, and in the interchange of information on administrative problems.

Mr. Sabin is maintaining, at the headquarters of the Cleveland Engineering Society and its affiliated societies, a complete list of standards approved by the A.E.S.C., and all information relating to the national standardization movement. It is hoped that arrangements will be made for him to sell copies of approved standards.

The Chicago Machine-Tool Meeting

THE Chicago Section of The American Society of Mechanical Engineers devoted the afternoon and evening of March 11, 1925, to a machine-tool meeting which was well attended. The program was designed to bring out the needs of machine-tool users and the advantages of different types of machines in finishing plane surfaces. The meeting proved to be interesting and stimulating, and encouraged the Chicago Section to announce that an annual machine-shop meeting would be held from now on.

James D. Cunningham, Chairman of the Chicago Section, presided at the session in the afternoon at which three papers were presented. The Trend of Machine Tool Design from the User's Standpoint was the subject of remarks by J. R. Shea.[1] Mr. Shea pointed out that in recent years the problem of designing articles for manufacture and the development of machine-tool equipment has been complicated by, first, improvements in raw materials possessing a wide range of physical characteristics, and secondly, by improvements in manufacturing processes. As an illustration of this Mr. Shea demonstrated how one particular part might be made by six different processes, and the relation of these processes to the development and use of machine tools. He discussed the transmitter face which is a part of the familiar desk telephone, explained how this part could be made from cast brass by an assembly of three punchings, by pressing from one piece, by hot pressing from brass, by molding from phenol plastic compound, and by die casting. Present practice was to manufacture the part from one piece of sheet brass.

Trend of Machine-Tool Design from the User's Standpoint

He said that machine tools selected or developed for the manufacture of a product should depend largely on the cost of the operation involved, the annual demand for the product, and the permanency of its design. General-purpose machines should be selected when the time consumed or quantity was small. Special machine-tool equipment should only be taken up if several different types of machines were running continuously on the same part.

In discussing the punch press, Mr. Shea stated that a large degree of safety might be obtained by proper tool design. Special pitmans with shear rings should be provided so that when the press was overloaded the rings would be cut and the pitman would telescope. Mr. Shea illustrated how increased production had been obtained by combining several operations in the same press. He stated that equipping screw machines and milling machines with individual motor drive was a distinct improvement in these types of equipment. Ball-bearing spindles had increased production and bettered the quality of work in drill presses. Automatic drilling machines offered a great possibility for reducing labor cost.

In the discussion the advantages of an individual motor drive were brought out and the importance of not using machine-tool equipment beyond the capacity for which it was designed was emphasized.

Characteristics Required in Machine Tools for Intensive Production

What We Want in Machine Tools was the topic assigned to Robert R. Keith,[2] who based his remarks on his experience in the manufacture of tractors. He first expressed the opinion that machine tools should be crowded to their comfortable capacity or even greater because machine tools became obsolete, and if the machines were used to the limit while they lasted, their users would be able to discard them in favor of machines of more recent design and take advantage of savings which the new machines offered. The rapid increase of intensive production had brought about specialization in laying out a shop for production, and lack of standard table height between various makes of machines often made it necessary that holes be dug in the floor so that working levels would be alike. Mr. Keith suggested the possible use of a primary beveled or spiral gear drive so arranged that the carrying arm for the pulley was on a swivel, which would permit its being turned to any angle that might be required in laying out the shop according to any particular scheme. Operating difficulties of machine tools, Mr. Keith said, might arise from two sources. First, from inadequacy of lubrication arrangements, and, secondly, from weaknesses in design. He especially criticized the practice of using plugs to cover oil holes. He also criticized the fact that repair parts furnished by machine-tool builders in many cases did not fit. Mr. Keith stated that he preferred to make his own repair parts and that he could make them more accurately to fit and of better materials. Touching on the common defects, he suggested that locking devices or adjusting rings could be improved and that loose-pulley bearings were inadequate. He also called attention to the fact that bearing sizes might be increased to take care of the strains caused by the operation of high-speed steel tools under high feeds when dull. He pointed out that in cutting cast iron which has a tendency to chatter, a weakness was generally revealed when torque strains were carried through long shafts or spindles. He also criticized the accuracy of the gears furnished on machine tools and directed attention to the need for adequate supporting tables on drill presses and greater rigidity of knee-type machines. He suggested the use of herringbone gears and flywheels to reduce chatter. In closing, he made a plea for more thorough attention to design of the product so that the amount of metal to be machined could be reduced by bettering rolling-mill, forge, and foundry practice.

Interchangeability of Repair Parts

The matter of interchangeability of repair parts was given a great deal of consideration in the discussion. It was explained that machine-tool builders must manufacture in small quantities. J. B. Armitage[1] pointed out that weaknesses of knee construction were easily overcome by bracing the overarm of the milling machine to the knee. He pointed out that it was common practice to use arbors that were too small, with the result that deflections were large and chatter prevalent. Mr. Armitage explained that frequent changes in design made it difficult for machine-tool builders to manufacture in large lots, which in turn necessitated a minimum expenditure for jigs and fixtures. This procedure of manufacturing in small quantities with frequent changes in design also made the provision of repair parts a serious problem. He pointed out that in answer to the suggestion that larger bearings be provided, that the cutter load was something over which the machine designer had no control. The machine would be designed for a certain horsepower and if a motor of that capacity was provided, the torque strains would not be excessive.

Die-Cast vs. Machined Parts

Die-Cast vs. Machined Parts was the subject of the paper by S. A. Hellings,[2] who described the method used in the die-casting process, stated the properties that might be expected in die-casting alloys, and showed the savings that might be obtained by using die castings instead of sand castings or machined parts. Mr. Hellings emphasized the recent advances made in casting aluminum-base alloys with one or more per cent of copper, silicon, zinc, nickel or manganese, and magnesium in varying proportions. A good die-casting alloy of this class consisted of $2\frac{1}{2}$ per cent copper, 3 per cent nickel and the remainder aluminum. Aluminum alloys could be made with an ultimate tensile strength ranging from 18,000 to 30,000 lb. per sq. in. with an elongation of $\frac{1}{2}$ to 5 per cent, and a Brinell hardness of from 45 to 100. Test bars $\frac{1}{2}$ in. by $\frac{5}{32}$ in. would bend from 12 deg. to 180 deg. and support a load on 3-in. centers of from 100 to 200 lb. Their specific gravity ranged from 2.60 to 3.10. These properties were in the cast condition. The heat treatment of aluminum castings of compositions corresponding to duralumin would increase the strength above these figures, but so far this heat treatment had been applied mainly to wrought aluminum or to special sand-cast parts. This

[1] Asst. Supt. of Development, Western Electric Company. Assoc. Mem. A.S.M.E.

[2] Chief Engr., Motor Trucks and Buses, International Harvester Co. Mem. A.S.M.E.

[1] Chief Engr., Kearney & Trecker Corp. Mem. A.S.M.E.

[2] Vice-Pres., Stewart Mfg. Co.

was because the process of die casting gave a fine, strong metal structure due to the chilling effect of the die, and consequently die castings were always stronger than the same alloy cast in sand. Aluminum alloys took and retained a high polish and consequently were used for parts which must be ornamental as well as serviceable.

In closing, Mr. Hellings expressed the hope that some time in the future bronzes, brasses, and possibly even iron and steel would be die-cast. They were now being cast in permanent molds.

The evening session of the machine-tool meeting was held at the City Club, following a dinner which was presided over by Mr. Cunningham. The first speaker was A. W. Robinson, one of the pioneer members of the Society, recently returned from Egypt and China. He recalled the early meetings of the Society and paid a tribute to its growth and influence. He pointed out the responsibility of the engineering profession in seeking not only the efficiency of machines, but the efficiency and well-being of men.

The technical session which followed the dinner was presided over by Hugo Diemer, Chairman of the Chicago Machine Tool Committee. The program consisted of the presentation of various methods of finishing plane surfaces. Large surfaces were treated from the point of view of heavy millers and planers; small surfaces from that of the milling machine and the grinder.

Heavy General Purpose Milling Machines

In his paper on Heavy Milling Machines, A. H. Lyon[1] paid particular attention to general-purpose machines such as those used for milling heavy castings in machine-tool shops, steel forgings in locomotive shops, and heavy steel castings such as safe-deposit-vault doors. He described the various types of milling machines and indicated their range of speed and power. He showed that the weight, power, and general usefulness had almost doubled in the past ten years. Heavy steel castings formerly considered as planer jobs were now milled successfully with decreased machine time. Locomotive side-rod forgings of alloyed steel might be slab-milled, two rods side by side on the table, at a feed of $8^7/_8$ in. per min. and might be channeled at a rate of 8 in. per min. Steel billets might have surface imperfections removed at a feed of 34 in. per min. using a 100-hp. drive motor, a 15-hp. table-feed motor, and a 10-hp. rail-feed motor. Cast-iron milling machine tables could be milled at the rate of 154 cu. in. of metal removed per min. The first cut for the T-slot in the table, seven slots each $^{29}/_{32}$ in. wide and $1^{15}/_{16}$ in. deep, were milled at the rate of 9 in. feed per min. Milling-cutter development had kept pace with the improvements in milling machines. Inserted-blade cutters were commonly used for roughing. Finishing cutters for cast iron were of high-speed steel and might be run at a speed of from 75 to 110 ft. per min. Holding fixtures were indispensable to single-purpose machines, but simple devices could be designed for use with several types of pieces and hold adequately.

In comparing milling machines with the planer, Mr. Lyon pointed out that in the milling machine the cutters had many teeth, most of which were in action at the same time, and with the multiple-head machine several cutters might be working at once. This accounted for the increased production that might be expected from the miller over that from a planer. Accuracy of machining reduced assembly cost in automobile work and the milling machine was the economical method of securing accurate results.

Planers for Large Surfaces

In the introduction to his paper, F. E. Cardullo,[2] who spoke on Planers for Large Surfaces, emphasized the fact that the planer was particularly adapted to work on cast iron and excelled in the production of accurate flat surfaces. As an example he presented the problem of planing a planer table, the casting weighing $8^1/_2$ tons and being 18 ft. long, 56 in. wide, and 11 in. in maximum depth. All surfaces must be parallel to within small limits and some must be accurate in dimensions to 0.00025 in. For instance, the distance between the V-centers, which was nominally 39 in., must not deviate from this value 0.0001 in. plus or minus. The included angle of the V must be correct within 20 seconds of arc, and the angle included between the plane through the V-surface intersections and the V-surfaces, must be correct to within 10 seconds of arc.

To set the table up and rough it out, removing about 2400 lb. of metal, took about 39 hr., floor-to-floor time. On the heavy roughing cuts metal was removed in a belt-driven machine at the rate of 40 cu. in. per min., or 600 lb. per hr., in spite of the fact that the tools were only cutting 75 per cent of the time.

The accuracy of the work done in this finished planing depended, first, upon the accuracy of the planer employed, secondly, upon the care with which the work was supported, stopped, and clamped, thirdly, upon the design of the gages employed, fourthly, upon the forms of the cutting tools used, and fifthly, upon the skill and experience of the operator. The time required for finished planing was 19 hr.

A planer-type milling machine of special design and extra heavy construction could rough this table out in less time than the planer. However, the cost per piece would be higher on account of the higher overhead, and the very much higher tool-room expense. Furthermore, the set up time for the milling machine would be much higher than for the planer, and the flexibility of the planer was much greater. It would, of course, be impossible for a milling machine to finish the work to the same accuracy, and it was doubtful if a milling machine could finish the job with sufficient accuracy to permit hand scraping to a fit.

There was one type of planer work where accuracy was not important, the only requirement being that the planer should hog off huge chips from the hardest and toughest steel that it was possible to machine. In such a machine designed to plane steel rails weighing 80 to 135 lb. per yd. and of high carbon and manganese content, cuts as heavy as $^1/_8$ in. deep and 3 in. wide were taken in two rails simultaneously at cutting speeds of 25 to 28 ft. per min. Such a planer could remove metal which was hard and tough enough to destroy milling cutters at the rate of 63 lb. per min., or, allowing for return stroke, reversal, and 10 per cent overrun, at the rate of 2700 lb. per hr.

Production possibilities of the planer, said Mr. Cardullo, depended upon fixtures, gages, tools, and methods of work. Tooling a planer for a production job was not as expensive as tooling a large milling machine. A great deal of time could be saved if the work could be gaged, and the gaging and setting of the tools needed only to be done once in order to finish five or even twenty pieces to the required limits.

The production from the planer varied greatly if the tools were not properly ground and with clearances and cutting angles suitable to the material to be cut. Tool shanks with inserted bits were often desirable. Special tool holders in which many bits might be inserted to work on a number of surfaces simultaneously were a great aid to production.

In closing, Mr. Cardullo emphasized the fact that the modern planer that had been developed during the last five years and would produce 30 to 60 per cent more work in the same time than the old type of planer. The modern planer properly tooled and handled by a skilled operator would produce work of great accuracy. It was a powerful production tool whose possibilities merited careful study by the production engineer.

Finishing Small Surfaces in the Milling Machine

The discussion on the finishing of small surfaces was opened by W. W. Tangeman,[1] who presented the case of the milling machine. After an introductory statement in which he gave a general description of the milling machine, Mr. Tangeman discussed the costs of machining operations. He separated the elements of costs of machining into power, perishable-tool cost and maintenance, interest on investment on machines and fixtures, machine maintenance, floor-space rental, and direct labor. He pointed out that power costs had been greatly reduced by the development of wide-spaced cutters combined with the use of proper rake angles, spiral angles, and tooth shapes. Whereas fifteen years ago the milling machine would remove approximately 3 to 4 cu. in. of cast iron per hp. per min., modern machines with modern cutters would remove from 1 to 2 cu. in. per hp. per min. on average roughing operations. The factor commonly used for steel was from $^1/_2$ to 1 cu. in. per hp. per min. and for softer metals such as brass

[1] Chief Engr., Ingersoll Milling Machine Co.
[2] Chief Engr., G. A. Gray Co. Mem. A.S.M.E.

[1] Sales Mgr., Cincinnati Milling Machine Co. Mem. A.S.M.E.

aluminum, etc. its value would run as high as from 4 to 6. As it cost from $25 to $30 per hp-year to operate a machine tool, power consumption was an important item to consider when deciding on a proper machining method.

Mr. Tangeman presented a simple formula used for figuring the feed in a given milling operation, namely,

$$\text{Feed per min.} = \frac{H \times K}{A}$$

where H was the horsepower input of the milling machine, K a constant depending upon the metal being milled, and A the area of the cut. He presented values for K as follows:

Cast iron	1.00	Brass	2.0 to 3.5
Steel	0.50	Aluminum	4.0 to 6.0
Malleable iron	0.75		

To indicate the item of perishable-tool cost Mr. Tangeman presented figures for an inserted-blade cutter and for a plain spiral mill. He stated that it was common practice to use solid face mills up to $3^1/_2$ in. in diameter and to use inserted-blade cutters above $3^1/_2$ in. In rough-milling vise bodies, removing approximately $^1/_4$ in. of metal, he gave the initial cost of the $8^1/_2$-in-diameter face mill as $90. The 12 blades inserted cost $36. Allowing 85 grindings for each set of blades the blade cost per grind was 43 cents. The cost of grinding was given as 38 cents and a body cost per grind 13 cents. The total cost for the grind was 94 cents. At a feed of $7^1/_4$ in. per min. 200 pieces could be milled per grinding at a cost per piece for the milling cutter of 0.0047 cent. The metal removed from the pieces milled was approximately 2300 cu. in. or 600 lb. of cast iron at a cost for the milling cutter of 94 cents.

In the second example, using a plain spiral mill working on cast-iron milling-machine gibs, he found that the total cost per grind was $1. In the intervals between grinding 150 pieces were cut making the cost for the metal 0.0066 cent per piece. At a feed rate of $18^3/_{16}$ in. per min. 1523 cu. in. of cast iron or 397 lb. were removed per grinding of the metal at a cost of $1 for the perishable tool cost.

In discussing interest on investment in machine and fixtures, Mr. Tangeman stated that the cost would be no greater for a milling machine than for an equivalent grinding machine. For maintenance, milling machines show a lower cost than grinding machines because of the lower operating speed and the fact that the moving parts were not subjected to the wash of abrasives as in grinding machines.

He pointed out that direct labor cost could not be determined on a theoretical basis but would depend upon the annual requirements of the part under consideration and the possibilities of tooling for maximum production. During the remainder of his talk the author illustrated many different types of milling operation with the production methods adopted.

Flat-Surface Grinding Machines

The Problem of Disk Grinders or Flat-Surface Grinders was taken up by F. E. Gardner.[1] After a brief review of the history of flat-surface-grinder development and methods of using abrasives, Mr. Gardner showed a series of slides which illustrated the production of plain surfaces by high-speed grinding. He emphasized the fact that rigidity and ample power were highly essential to successful grinding-machine operation. He pointed out that disk-grinding operations would cost less and better results would be produced if the surface to be ground was recessed or relieved to reduce the amount of material that must be removed in the grinding. He stated that the field of the grinding machine was where light, rapid cuts were required, sometimes in sand, sometimes in scale, where the fixtures or holding devices might be simple, and where accuracy was required. In several of his slides Mr. Gardner illustrated the possibilities of having greater production with less effort on the part of the operator by automatic feeding devices.

After the formal papers, the meeting broke up into small groups for the informal discussion of the points brought out by the speakers and for the inspection of many samples of different classes of workmanship from the various machines described.

[1] Vice-Pres., Gardner Machine Co.

Hydroelectric Symposium at Philadelphia

THE third annual hydroelectric conference was held at Philadelphia on March 10, 1925, under the auspices of the Engineers' Club of Philadelphia. The morning and afternoon sessions were held in the auditorium of the Manufacturers' Club and the evening session and dinner in the ball room of the Bellevue-Stratford.

The morning session was opened by an introductory address by Conrad N. Lauer (Mem. A.S.M.E.), President of the Engineers' Club, and the technical part was carried on with Dr. Arthur M. Greene, Jr. (Mem. A.S.M.E.), in the chair.

The first paper was by Wm. M. White (Mem. A.S.M.E.), on the Operation of Hydroelectric Systems with Auxiliary Steam Plants for Best Economy and Proper Governing. After giving an outline of the fundamentals of hydroelectric-unit operation in its interrelation to the operation of steam plants, the author pointed out that, contrary to the popular notion, it was often more economical during the low-water season to operate the steam plant on base load and take the peak of the load of the hydroelectric plant. Relative efficiencies of hydroelectric units were discussed and the fallacy of blocking a number of hydroelectric units at one point and operating others at part load for governing was emphasized. The author also suggested specifications which should control the design of the governor of steam auxiliary units for most economical operation with hydroelectric plants, expressing the belief that a shorter time period of operation should be given to the steam units. This point provoked objections in the discussion later on.

Forrest Nagler (Mem. A.S.M.E.), in a paper entitled Operation of Hydroelectric Units for Maximum Kilowatt-Hours, pointed out the magnitude of certain losses that commonly exist in hydroelectric units under conditions of average operation and outlined a method for reducing such losses in order to further the primary object of securing the maximum number of kilowatt-hours from the station. Comment was made particularly on the extent of losses in turbines and impulse wheels under conditions of synchronous-condenser operation of the units. The possibility of eliminating or greatly reducing these losses by controlled venting of the runners was outlined. Losses inherent in impulse wheels were isolated with a view to pointing out the possibilities in their reduction. Typical efficiency curves were shown for illustrating the extent of gains possible, and devices were shown or described for accomplishing these results.

Frank H. Rogers (Mem. A.S.M.E.), and Lewis F. Moody (Mem. A.S.M.E.), presented a paper entitled Interrelation of Operation and Design of Hydraulic Turbines. Among other things, they contended that frequently turbines of high speed were chosen without proper regard for stream flow and part-load operation. The greater part of the paper was devoted, however, to the presentation of a method of determining the proper division of various flows between units in a plant so as to obtain maximum output for each flow. Output-discharge curves of different types were considered and a general method proposed applicable to any particular installation having the shape of the output-discharge curves of the units determined.

The selection of type of turbine to avoid conditions conducive to pitting or corrosion was studied from a new viewpoint. Instead of selecting speed from an "experience curve" plotted between specific speed and head, it was proposed that the selection be governed by reference to the minimum absolute pressure to be expected at a local point within the turbine. The drop in pressure below atmospheric was expressed in the sum of static draft head, velocity head regained in draft tube, and a local pressure drop proportional to the head on the turbine. Effects of variation in operating conditions such as overload operation and variation of tailwater level were considered.

General information on hydroelectric developments on the Pacific coast was presented in a paper by Ely C. Hutchinson (Mem. A.S.M.E.). This paper contained particulars regarding the impulse or Pelton water-wheel construction extensively used in the West, as well as a description of the high-head turbine installations on the systems of the Southern California Edison Co., and the Portland Electric Power Co., including the Pit River No. 3 power-

house of the Pacific Gas and Electric Co. with its three 33,000-hp. Pelton turbines.

The afternoon session, devoted to the general subject of electrical machinery, was held with Robt. H. Fernald (Mem. A.S.M.E.) in the chair. The first paper was by Chas. B. Hawley and dealt with the operating practice in hydroelectric developments, and more particularly with plant efficiency, practical operating conditions, emergency equipment and protective devices.

N. C. Olson in a paper entitled Feature of Design in Latest Large Hydroelectric Generators, gave data regarding present hydroelectric developments and future possibilities, and described some of the world's largest generators, such as the 65,000-kva. machines of the Niagara Falls Power Co., the 32,500-kva. generators for Muscle Shoals, etc. Among other things, an interesting description of the so-called "one-man" station of the Minnesota Power and Light Co., was given, including its overhung-rotor vertical generator with a capacity of 15,000 kva. From a curve presented by the author based on the sales of the General Electric Co., it would appear that while horizontal generators were predominant four or five years ago, the tendency is rapidly veering in favor of the vertical type.

F. D. Newbury (Recent Developments in Hydroelectric Generators) stated that an outstanding characteristic of the hydraulic-turbine generator was its diversity of form. It must meet such a wide variety of conditions that generators of this class were rarely duplicated.

In spite of this situation, there was still a basic design that was generally followed by the designers of this class of machinery. The mechanical specifications of this basic design were listed by Mr. Newbury and recent developments were discussed under the headings of rotor construction, accessibility of parts, fire protection, and maximum possible ratings.

At the evening session Dr. Wm. F. Durand, President of The American Society of Mechanical Engineers, gave an address which showed the necessity for conserving oil to postpone the threatened exhaustion of the existing supply in about twenty years. He also dwelt upon the supremacy of coal in the production of power and the necessity of supplementing water power with steam plants.

Charles Penrose (Mem. A.S.M.E.) attacked the giant power scheme for Pennsylvania from legal, economic, and technical standpoints. Some of his objections to it were as follows: Because of the fact that the commercial functioning of the giant power depends on the ability to dispose of by-products of coal, it would be far more subject to fluctuations in economic conditions in industry than were the power companies of today with their highly diversified load. Lack of water for operating the condenser equipment of mine-mouth plants would either make these plants dangerously dependent on seasonal variations of rainfall or would force them to operate non-condensing, and this latter was notoriously wasteful of fuel. There was not sufficient experience with cooling-tower operation to permit reliance on it for such enormous units as were proposed under the giant power scheme. Because of line and plant-trouble possibilities the giant-power operation would be, in general, less reliable than the present smaller scattered generating plants. It would be therefore comparatively unattractive to industries such as textile, chemical, electric furnaces, etc. where even short interruptions of service would be very harmful.

A.I.M.E. Annual Meeting

THE 131st Annual Meeting of the American Institute of Mining and Metallurgical Engineers held in New York from February 16 through 20, 1925, was featured by a number of papers and addresses of interest to mechanical engineers.

The annual lecture delivered at the Institute of Metals Division was given by Dr. Carl Benedicks, Director of the Metallographic Institute of Stockholm, on Certain Factors Influencing the Rapid Corrosion of Boiler and Condenser Tubes. In the same section was given a paper by Robt. S. Archer and Zay Jeffries entitled New Developments in High-Strength Aluminum Alloys. In this paper two new alloys of the "strong" alloy class having improved fabricating qualities were described; also the methods of producing alloys of the duralumin type with greater strength and hardness than previously obtained were given. Malleability of Nickel was discussed by Paul D. Merica and R. G. Waltenberg, who showed that lack of malleability in ordinary nickel was due primarily to the presence of sulphur in amounts as low as 0.005 per cent and the elimination of this sulphur made nickel malleable. Recrystallization and Grain Growth in Soft Metals such as Lead, Tin and Cadmium formed the subject of a paper by Maurice Cook and Ulick R. Evans.

In the section dealing with coal special attention may be called to the paper by John W. Cobb on Nitrogenous Constituents of Coal and the three papers by R. V. Wheeler and his collaborator on the Constitution of Coal and Other Developments in the Chemistry of the Master Fuel Resulting from Research in England.

In the Iron and Steel Division the Howe Memorial Lecture was given by Dr. John A. Mathews on the subject of Austenitic Steel, as well as several papers on metallurgy, including the subject of reversing-rolling-mill practice.

Giant Power

IN FEBRUARY, the Giant Power Survey Board submitted its report to the General Assembly of the Commonwealth of Pennsylvania. In transmitting it, Governor Gifford Pinchot, who served as Chairman of the Board, outlined the following essentials of cheap and abundant power, legislation for which is urged by the Board:

1 Mass production, with opportunity for by-product recovery—to be secured by generating stations of 300,000 kw. or more in or near the coal fields supplying transmission lines of 110,000 volts or more connecting with all other major transmission lines in the state.

2 The creation of a common pool of power into which current from all sources will be poured, and out of which current for all uses may be taken—to be secured by organizing giant power companies to purchase surplus power from all generating stations in the state and sell to all distributing systems in the state.

3 Free access by every water-power and steam generating station to every distributing system in the state which supplies the consumer—to be secured by making all major transmission lines common carriers of current from the giant power companies and other generating stations to any and all distributing systems in the state.

4 Complete, prompt, and effective regulation of rates, service, and security issues—to be secured by fundamental changes in the Public-Service-Company Law.

5 Rescue of the regulation of electrical service from the destruction now threatened by its conversion into interstate commerce, which will be beyond the control of the states and has not been regulated by Congress—to be secured by compacts among the states consented to by Congress, as allowed by the Constitution of the United States, or failing that, by Congressional legislation.

6 Systematic extension of service lines throughout the rural districts.

Morris Llewelyn Cooke served as Director of the Survey on which the technical reports are based.

Oil and Gas Power Week

THE program for the country-wide celebration of Oil and Gas Power Week from April 20 to 25 is rapidly assuming tremendous proportions. Over eighty meetings of the sections of participating societies of technicians and engineers are to be held, at which the petroleum situation is to be reviewed from all angles. The efficient utilization of the limited supply of an important fuel is a subject that affects the entire engineering profession, and its treatment in this inclusive manner by fifteen technical bodies in close coöperation insures the arousing of widespread public interest in the discussion.

The fifteen national bodies that are participating in the activities of Oil and Gas Power Week, which will reach eighty centers, are providing publicity for the programs of these local meetings.

A Correction

ON page 29 of the January issue of MECHANICAL ENGINEERING in Mr. Wolff's paper on Test of Pulverized-Fuel-Fired Boilers at the Lake Shore Station, Cleveland, a sentence reads, "The draft loss shown in Table 2 is therefore not representative." This should have read, "The draft loss shown in Table 1 is therefore not representative."

Book Reviews and Library Notes

THE Library is a cooperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

The Elements of Machine Design

ELEMENTS OF MACHINE DESIGN. By S. J. Berard and E. O. Waters. D. Van Nostrand Co., New York, 1924. Cloth, 6 × 9 in., 323 pp., illus., diagrams, tables, $2.50.

THE criticism is often made that the teaching in the technical schools, particularly in mechanical engineering, always seems to lag several years behind the progress of actual conditions in the industry. This is strikingly apparent in the recent book on Elements of Machine Design by Berard and Waters where it touches on standard machine elements, such as screw threads, and the subject of manufacturing tolerances. These glaring anachronisms almost spoil an otherwise very able work on this subject.

In general, the choice of material is good, its arrangement excellent, and its presentation clear, and precise. In regard to the material on screw threads, it is evident that the authors wrote without any recent first-hand knowledge of the subject. In fact, there has been a very marked development in machine design in the industry during the past few years that the authors have ignored entirely. This is the factor of standardization. Long before general industrial standards of mechanical elements are possible, individual plants develop certain standards of construction for their own use. These standards cover a very wide field, only a very small proportion of which have been considered as general engineering standards. This subject is a very vital one as regards economy of manufacture, even though individual special machines are to be constructed. A few pages at least in such a text as this should have been devoted to this topic to give the student at least a warning that this feature must be considered seriously in actual industrial work.

The material on screw threads evidently was taken from old handbooks and reports. A very complete standardization and simplification of screw-thread sizes[1] has been before the engineering public in the form of printed reports, abstracts in all of the technical papers, etc. for a considerable time prior to the publication of this volume. The later editions of handbooks on mechanical engineering also carry this revised and up-to-date information. This oversight of the authors is regrettable.

EARLE BUCKINGHAM.[2]

Books Received in the Library

CARBURANTS NATIONAUX, Exposition de Buc-Congrès; Compte Rendu des Travaux du Comité Central de Culture Mécanique en 1924. Chaleur et Industrie, Paris. Paper, 6 × 9 in., 264 pp., 15 fr.

As a supplement to its December, 1924 issue, *Chaleur et Industrie* publishes the proceedings of the Congrès des Carburants held in Paris, in October, 1924, under the auspices of the Ministry of Agriculture. The specific problem that occupied the attention of the Congress was the transformation of mineral and vegetable substances of French origin into fuels for internal-combustion engines. Among the subjects discussed were the Berthelot process of hydrogenation, the Bergius process of liquid-fuel production, the manufacture of synthetic petroleum from vegetable and animal oils, the theory underlying synthetic motor fuels, the alcohol engine, the production of motor fuel by the carbonization of solid fuels, the use of vegetable oils, producer-gas trucks and farm tractors, colloidal fuels, and the use of power in farming.

CHARTS AND GRAPHS. By Karl G. Karsten. Prentice-Hall, Inc., New York, 1923. Fabrikoid, 6 × 9 in., 724 pp., illus., diagrams, charts, $6.

A comprehensive text on the theory and use of charts of all kinds, with instructions for making them. The author adopts a theory of the evolution of charts, in accordance with which all forms fall into line with simple origins and clear channels of growth, and his text follows this evolution, proceeding from simple, non-mathematical forms to complicated ones. The proper uses of each form and the purposes to which it is suited are discussed thoroughly.

COAL CATALOG combined with COAL FIELD DIRECTORY. 1925. Keystone Consolidated Publishing Co., Pittsburgh, 1925. Fabrikoid, 9 × 11 in., 1302 pp., illus., maps, $15.

The 1925 edition follows the lines of previous ones, but numerous refinements have been added. Among these are quantity analyses of delivered coal mined in various states; additional maps covering Kentucky, Virginia, and Maryland completely, and two new districts in Illinois; and an alphabetical list of all mines east of the Mississippi River. The Catalog is a most useful directory to the coal miner and purchaser, making it possible to answer almost every question that arises concerning the coal resources of a region, the quality of a coal, the location of a mine, or the resources of a mining company.

ECONOMICS OF BUSINESS CYCLES. By Arthur B. Adams. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 268 pp., $2.50.

In this book, the Dean of the School of Business at the University of Oklahoma presents an analytical study of the causes which produce the different changes in the trend of business and of the ways in which these forces may be controlled to eliminate violent fluctuations. As a preliminary to his discussion of business cycles, he also gives an analysis of the nature and characteristics of the modern business system. The book is intended primarily for college class use, but will also prove useful, the author hopes, to business men.

FÜNFUNDZWANZIG JAHRE ZEPPELIN-LUFTSCHIFFBAU. By L. Dürr. V. D. I. Verlag, Berlin, 1925. Paper, 9 × 12 in., 83 pp., illus., diagrams, 8 gold marks.

The completion of the *Los Angeles* marked the end of twenty-five years' activity for the Zeppelin factory, an occasion that the company has commemorated by the present volume. The book contains a detailed description of the *Los Angeles* and also interesting accounts of the progressive changes in the design of the Zeppelin and of its structural parts which have occurred during these twenty-five years. These have much historic interest as a record of the way in which a practicable airship has been evolved.

HYDRAULICS, WITH WORKING TABLES. By E. S. Bellasis. Fourth edition. E. & F. N. Spon, London, 1924. Cloth, 6 × 9 in., 348 pp., illus., tables, 18 shillings.

The objects of this book are to deal thoroughly with the facts, laws, and principles of hydraulics, keeping always in view their practical aspects, and to provide accurate coefficients, based on recent research. Obsolete matter and needlessly long mathematical

[1] Progress Report of the National Screw Thread Commission (Jan., 1921); American Standard Screw Thread, A. E. S. C. (July, 1924).

[2] Engineer, Pratt & Whitney Co., Hartford, Conn. Mem. A.S.M.E.

investigations are avoided. New material has been added to this edition concerning recent work in connection with flow through sluices, with the laws governing silt and scour, and with new methods of discharge observation.

KÄLTEPROZESSE; Dargestellt mit Hilfe der Entropie-tafel. By P. Ostertag. Julius Springer, Berlin, 1924. Paper, 6 × 10 in., 118 pp., diagrams, plates, 6.80 gold marks.

This book is written for designers of refrigerating machinery, to whom it offers a rapid method for making the necessary calculations, based on the entropy diagram. The work is divided into four sections. The first discusses the thermodynamic principles involved and is accompanied by good entropy diagrams for ammonia-carbon dioxide, and methyl chloride. Section two treats of several modern refrigerating processes, especially multi-stage machines. A special section is devoted to heat conduction, in which calculations for a double-pipe condenser are carried out in detail, as an example. The fourth section deals with steam condensers and the fifth with the use of gases for the production of cold.

LES LIGNITES ET LEURS APPLICATIONS INDUSTRIELLES. By Edmond Marcotte. Gauthier-Villars et Cie., Paris, 1925. Paper, 5 × 8 in., 327 pp., illus., diagrams, 20 francs.

The primary purpose of this book is to arouse interest in the lignites of France and their possible utilization as a substitute for the imported fuel upon which that country now depends. The book discusses the available deposits, describes the processes that have proven most successful, in France and elsewhere, for mining, drying, and briquetting lignite; and also discusses its utilization by gasifying and low-temperature carbonization, and the recovery of by-products. Economic questions are considered as well as engineering ones.

MANUFACTURE OF PULP AND PAPER, vol. 5. By Joint Executive Committee of Vocational Education Committees of the Pulp and Paper Industry of the United States and Canada. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., various paging, illus., diagrams. plates, $5.

This volume completes the textbook on pulp and paper manufacture prepared with the support of the pulp and paper manufacturers of North America. The work covers the entire field, including the necessary elementary knowledge of mathematics and natural sciences, and is intended for self-instruction as well as for class use. Much of the information has not been easily accessible hitherto. The present volume deals with paper-making machines, hand-made papers, tub sizing, paper finishing, coated papers, and the testing of papers. A bibliography on testing is appended to the last subject. There is also a selection on general mill equipment, pumps, electrical machinery, ventilation, lubrication and water.

OFFICE MANAGEMENT; Principles and Practice. By William Henry Leffingwell. A. W. Shaw Co., Chicago and New York, 1925. Cloth, 6 × 8 in., 850 pp., illus., chart, $5.

A treatise on the organization, standardization, and control of office work. Mr. Leffingwell attempts to explain the scientific basis of office procedure and to show how the principles of scientific management may be applied to office work. He has not given a set of ready-made plans, forms and systems; instead he shows how to analyze the problems of an office and plan appropriate ways for meeting them efficiently.

PETROLEUM REGISTER. July, 1924. Oil Trade Journal, Inc., New York, 1924. Cloth, 9 × 12 in., 550 pp., maps, $10.

Covers all these engaged in the petroleum industry, as producers, refiners, marketers, manufacturers of natural gasoline, drilling contractors, geologists, equipment manufacturers, pipe-line companies, etc. Information is given concerning the officers, capitalization, and plants of the companies. Other features are a dictionary of trade names, a list of natural gasoline plants, a directory of oil associations, statistics of production, prices, stocks, etc. and outline maps of the producing fields in America and the Dutch East Indies.

STEAM POWER PLANT ENGINEERING. By G. F. Gebhardt. Sixth edition. John Wiley & Sons, New York, 1925. Cloth, 6 × 9 in., 1036 pp., illus., diagrams, tables, $6.

A comprehensive description of present American practice in steam power-plant equipment and operation. Engine and boiler design are omitted, except from the point of view of the operating engineer. Because of its completeness as a picture of current methods, the book should be useful as a handy book of reference as well as for use as a textbook.

TECHNICAL MECHANICS, STATICS, KINEMATICS, KINETICS. By Edward R. Maurer and Raymond J. Roark. Fifth edition. John Wiley & Sons, New York, 1925. Cloth, 6 × 9 in., 364 pp., diagrams, tables, $3.50.

This may be fairly described, the authors say, as a theoretical mechanics for students of engineering. It is not comparable to books commonly called Theoretical Mechanics, generally intended for students of mathematics or physics, nor to those commonly titled Applied Mechanics, which include the strength of materials, hydraulics, etc. The title of Technical Mechanics has been chosen to indicate this differentiation. This edition is rewritten, with improvements in the presentation, changes in subject matter and arrangement, and many new problems.

TIME MEASUREMENT. By L. Bolton. D. Van Nostrand Co., New York, 1924. Cloth, 5 × 8 in., 166 pp., illus., diagrams, $2.

A brief account of the rudiments of the measurement of time, in which are described the more prominent natural phenomena suitable for this purpose and the machines constructed for it. A good introduction to the subject for those who wish to know how clocks, watches, and electric clocks are made.

VORLESUNGEN ÜBER WASSERKRAFTMASCHINEN. By R. Camerer. Second edition, revised by Bernhard Esterer. Wilhelm Engelmann, Leipzig, 1924. Cloth, 8 × 11 in., 515 pp., illus., plates, diagrams, $6.72.

A comprehensive textbook on hydraulic turbines. The author commences with an explanation of the general principles underlying hydraulic technique and then sets forth those portions of hydraulics which the turbine engineer should understand. He surveys hydraulic motors, giving during this survey an introductory general description of turbines. The general theory of the turbine is developed, followed by a section on the theory of the Francis turbine. The sixth section treats extensively of the design and construction of Francis turbines. The book ends with a brief section on the design and construction of Pelton turbines. It is unusually well provided with fine plates.

WIRTSCHAFTLICHE VERWERTUNG DER BRENNSTOFFE. By G. De Grahl. Third edition. R. Oldenbourg, Münich & Berlin, 1923. Paper, 8 × 11 in., 649 pp., illus., diagrams, 32 marks.

An exhaustive treatise on the efficient utilization of fuel, written with particular reference to present conditions in Germany. Discusses the available fuels, methods of improving their quality (coking, gasifying, etc.), firing methods, heat economy and similar topics. Economic and engineering considerations are both covered.

DIE ZEITGEMÄSSE HEISSDAMPFLOKOMOTIVE. By Robert Garbe. Julius Springer, Berlin, 1924. Boards, 8 × 11 in., 167 pp., illus., diagrams, tables, $3.35.

This treatise discusses the development of the locomotive in recent times, with particular reference to the use of superheating. The author discusses the investigations on the relations between grate surfaces and heating surfaces, the combustion phenomena in the firebox, methods of firing, and the efficient production and utilization of superheated steam. From these investigations he draws conclusions concerning the design and operation of freight and passenger locomotives and illustrates his opinions by applying them to the design of an actual locomotive.

DIE ZUSTANDSGLEICHUNG VON GASEN UND FLUSSIGKEITEN. By J. J. Van Laar. Leopold Voss, Leipzig, 1924. Paper, 6 × 10 in., 368 pp., $3.45.

For over thirty years Dr. van Laar has been an active investigator of the phenomena accompanying the change from the liquid to the gaseous state, and of the equation of state. The results of his work, scattered through many periodicals, are now presented as an organic whole in the present volume, which should be of considerable interest to students of thermodynamics and physical chemistry. The book deals particularly with the variability of the constants of Van der Waal's equation, with critical conditions, vapor pressure phenomena and coexisting vapor and liquid phases.